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A SUPERCHARGED HEAVY DUTY GAS TURBINE AND STEAM CYCLE AS A PROPULSION PLANT FOR NAVAL AUXILIARY SHIPS

ROBERT BRUCE WOODRUFF

A SUPERCHARGED HEAVY DUTY GAS TURBINE AND STEAM CYCLE AS A PROPULSION PLANT FOR NAVAL AUXILIARY SHIPS

by

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Lieutenant Commander, United States Navy

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(1964)

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ROBERT BRUCE WOODRUFF

Submitted to the Department of Ocean Engineering on June 2, 1972, in partial fulfillment of the requirements for the degree of Ocean Engineer and Master of Science in Mechanical Engineering.

ABSTRACT

A large, heavy duty industrial gas turbine is proposed as the prime mover for the propulsion plant in a large displacement high-speed Naval Auxiliary ship of the Fast Combat Support Ship Class (AOE-1 Class). The analysis consists of showing the feasibility of such a plant when operating at full power, and includes size, weight and cost data along with a suggested plant layout for this specific class of ship.

The cycle is of the combined gas turbine and steam type, and consists of a General Electric frame size 5 industrial gas turbine. The inlet to the compressor of the gas turbine is supercharged to 1.4 atmospheres by a 2 stage axial compressor which is steam driven. An external intercooler using seawater removes the heat of supercharger compression, and thus the inlet temperature to the main compressor is the same as ambient. The higher pressure at inlet, coupled with no increase in temperature, allows the gas turbine to perform at significantly higher power levels. The ship is driven by a controllable pitch propeller at 50,000 SHP.

Exhaust from the gas turbine is used to generate steam in an unfired waste heat recovery steam generator. The steam pressure is at 450 psig, 750° F. and is used to drive the supercharger and for turbogenerator sets, as well as for normal hotel loads. This gives an all-purpose fuel rate at full power of 0.444 and a thermal efficiency of 0.316, base on Navy Distillate fuel.

Among the conclusions reached concerning the feasibility of placing a supercharged heavy duty gas turbine in a marine environment are the following:

1) It would be feasible to place a heavy-duty supercharged gas turbine and steam cycle aboard a large Naval Auxiliary ship.
2) The specific fuel rate at full power, 0.444, is significantly better than that of the presently installed steam plant and compares favorably with aircraft derivative marinized gas turbines. The annual saving in fuel costs to the Navy would be on the order of 30%.
3) While supercharging of the gas turbine is an interesting thermodynamic consideration, it is concluded that, in a marine propulsion plant with a combined cycle, the steam generated is best used by coupling a steam turbine directly to the main shaft.

4) The use of intercooling a gas turbine cycle aboard ship is particularly useful as it reduces sea salt concentrations.
5) The life cycle costs of a steam plant are less than that of a supercharged gas turbine cycle.

Thesis Supervisor: A. Douglas Carmichael

Title: Professor of Power Engineering

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Special thanks are extended to my wife, Ellen, for her understanding while I typed in the small hours of the morning--with a new baby in the house. She also proofread the final thesis draft prior to submission.

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1. INTRODUCTION

The recent developments in the use of gas turbines for propelling ships has produced significant speculation as to the direction they are actually going to take in the coming years. The idea is still too new, the machinery not yet fully proven, to be able to draw any significant conclusions about the future. The gas turbine does have the general advantages of ease in starting and automation, as well as low specific weight and volume. Some of the aspects which have tended to hold it back as a prime mover for ships are the fact that it is uni-directional (arrangements necessary to go astern), has poor performance except when at or near full power, and has a thermal efficiency which generally is lower than steam throughout the power ranges.

The U. S. Navy and other navies of the world, as well as a few merchant ship owners, have purchased and tested second generation marinized aircraft derivative gas turbines with a fair amount of success. (1, 2, 3, 4, 5) These engines have been developed from those used to power aircraft, and are small, very efficient, and operate at high pressure ratios. The other general family of gas turbine, the industrial, or heavy-duty type, has seen many hours throughout the world, primarily in land based requirements such as power generation, gas pipelines and chemical processing plants. They have received very little, if any, attention in consideration as a propulsion plant for large ships. In the late 1950's, the Maritime Administration undertook a program to look at different propulsion plants for merchant ships. One of these involved the use of an industrial gas turbine, and for four years the ship steamed successfully with very little maintenance problems, amassing a total of 10,000 hours

on the line with no major casualties. (6) The funds for this ship ran out in 1959, and there was no follow-on program to pursue the use of the heavy duty gas turbine in the shipboard environment.

The U. S. Navy in the 1960's continued to rely on steam plants for all classes and sizes of ships. The United States has enjoyed a very favorable position in steam technology since World War II--in design, manufacture, as well as operating experience. The late 1960's, however, showed some serious drawbacks to steam plants in the specific aspect of reliability and availability, and the numerous failures for the Navy's "ships of the line" began to wear severely the patience of its operational commanders. The early part of the decade had seen wide acceptance of aircraft derivative gas turbine for unconventional craft such as hydrofoils, patrol craft (7), and surface effect ships--any other power plant could not compete with its low pounds per horsepower ratio. Additionally, numerous studies had been made using combined cycles of steam and gas turbine (COSAG) (8), and many foreign navies had been using or building gas turbine plants since the first part of the decade. (9, 10, 11, 12) Thus the stage had been set for the gas turbine. With the status of the supercharged or pressure fired boiler as a maintenance disaster,* the fate of the steam plant was sealed. The success of the GTS Adm. William M. Callaghan in the use of aircraft derivative gas turbines convinced the U. S. Navy that there was indeed a future for this propulsion plant. The aforementioned advantages, plus the ability of an engine change out within

*The author had the opportunity to serve as Engineer Officer on one of these ships, the USS J.A. Furer, a guided missile escort destroyer.

hours (as opposed to any maintenance aboard ship) gave the plant impetus to cause the Navy to choose aircraft derivative gas turbines for the new Spruance Class destroyer (DD 963) and the Patrol Frigate (PF) Class.

The U. S. Coast Guard, in its Hamilton Class Cutter (WHEC 378 Class) uses Pratt and Whitney FT4-A aircraft derivative gas turbines coupled with diesels for cruise power. While all ships have proven to be fairly reliable, first indications are that the gas turbine plants may be extremely costly to maintain. Most of the ships have had at least one change out of an engine, and some have had two, at an estimated cost of 150,000 dollars per change. The average time on these engines has been between 2-3,000 running hours. While the engine change outs are relatively easy to accomplish in a short amount of time, there still remains a tremendous cost factor, and the "zip in, zip out" concept needs a second look from the dollars and cents point of view. (13) The second major problem with these engines has been failure of inlet guide vanes at about the 3,000-hour mark at an estimated cost of replacement of 20,000 dollars. In short, one must ask the rhetorical question of whether the aircraft derivative engine is truly suitable for twenty years in a marine environment where it must digest large quantities of salt air and where the machines are subjected to constant cyclic "G" forces in six degrees of freedom? This is a far cry from an aircraft which spends much of its time at a constant altitude, at the same power levels, and with nearly the same inlet conditions.

While the aircraft derivative may eventually rule for ships with smaller displacements (hydrofoils, destroyers, etc.) the industrial gas turbine may be desirable for ships that are not volume limited. Larger ships in the Navy have always used steam plants, and the higher operating

costs at higher specific fuel rates have been accepted as the price to pay for good reliability, availability and maintainability. The industrial gas turbine in combinations with a waste heat boiler, or heat recovery steam generator, has thermal efficiencies and fuel rates significantly better than the steam plant.

The advantages of heavy duty gas turbines have been made well known by the manufacturers, who claim:

1) First cost competitive with A/C derivative G.T., diesel and steam;
2) High proven reliability in marine environment (SS John Sergeant) and nearly 35 million hours (GE) on the line--some of which has been in salt air environment in offshore drilling rigs;
3) More rugged construction than A/C derivatives with durability approaching steam turbines;
4) Longer times between overhaul than A/C derivatives (20,000 vs 4,000 hours) and lower maintenance costs than A/C G.T.;
5) Manning requirements (ease of automation) competitive with diesel and A/C G.T., easier to automate than steam;
6) Wider range of fuels allowed--although Navy distillate expected, unsettled world conditions may make it strategically important for support ships to be able to burn residual with minor modifications;
7) Lower lube oil costs;
8) Fuel rate competitive with diesel, steam and A/C derivatives;

9) Very low specific weight and volume for greater cargo capacity;

10) Shorter time to get on the line and flexible power range with two-shaft G.T.

With these factors in mind, coupled with the idea that industrial gas turbine will no doubt become more efficient as temperatures and pressure ratios increase, (Fig.1.1)(14) it appears that Navy support ships (or other ships with very large displacements) could make use of the industrial gas turbine as they operate at high speeds a larger portion of their time. The platform used in the analysis of this supercharged combined cycle is the Fast Combat Support Ship (AOE-1 Class) whose mission is to run with a Fast Carrier Task Force and keep it supplied at sea with a minimum of time away from operations.

A photograph of the ship Sacramento (AOE-1) is shown in Figure 1.2 (15). The Sacramento Class has the following general characteristics: (16)

Length: 792 feet

Beam: 107 feet

Draft: 39 feet

Displacement: 53,600 tons

Speed: 26+ knots

Shaft horsepower: Twin screw @ 50,000 SHP each

Crew: 608 officers and men

Fuel capacity: 8,000,000 gallons

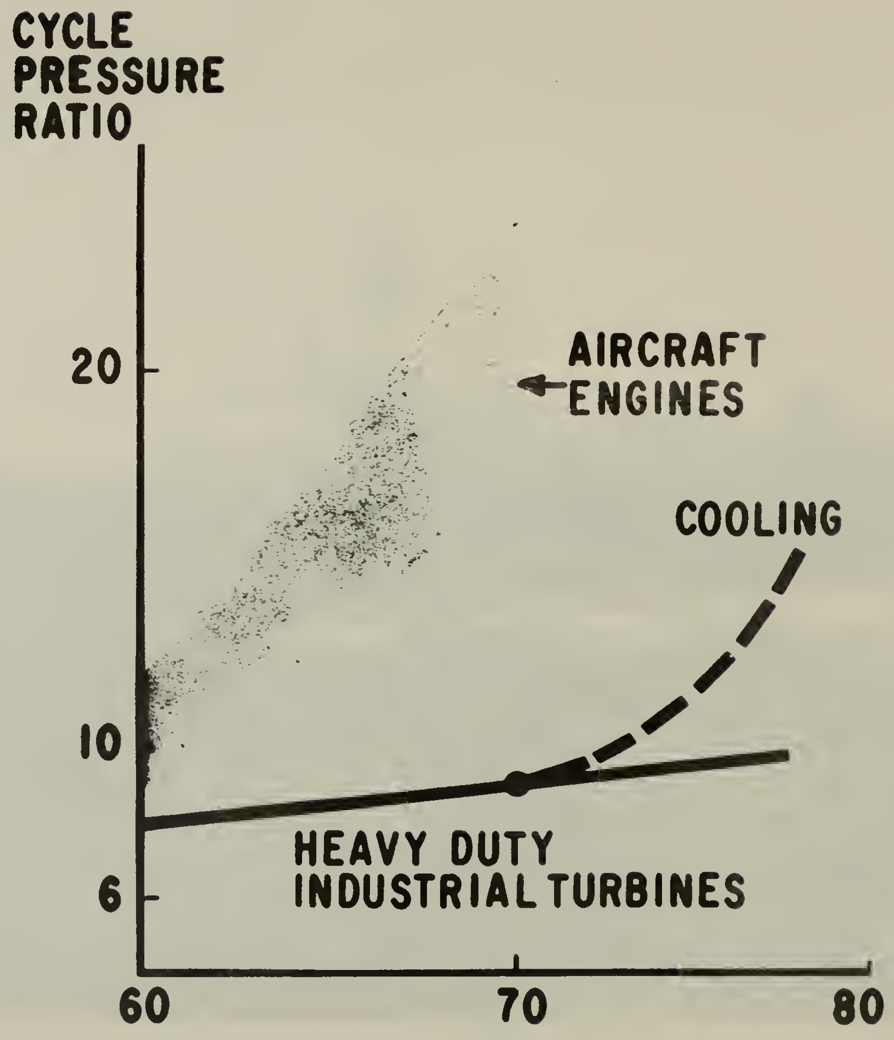


Figure 1.1 Gas Turbine Cycle Pressure Ratio History and Trends Through 1980 (14)

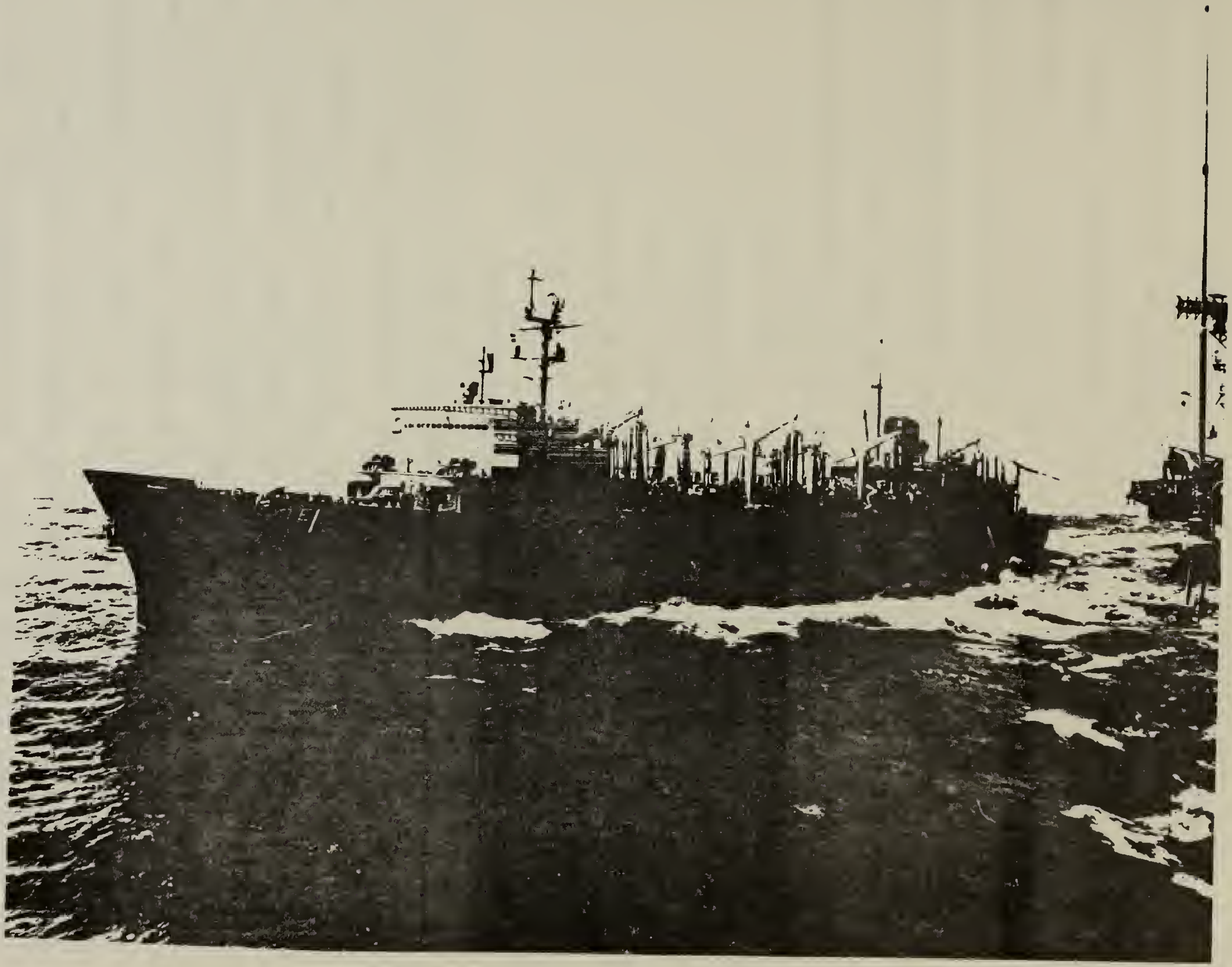


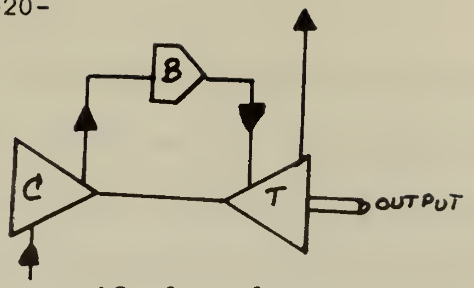
Figure 1.2 USS Sacramento (AOE-1) Underway (15)

2. SELECTION OF A GAS TURBINE CYCLE

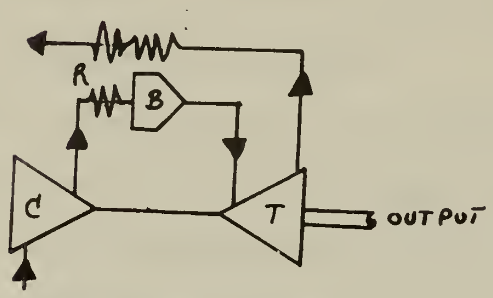
In Chapter 1, the feasibility of using an industrial gas turbine as a prime mover for ships was discussed. As stated earlier, most of the heavy-duty gas turbine applications would be expected to be used in a larger ship, on the order 20,000 tons or greater. From a thermodynamic standpoint, what would be the most effective cycle to use to drive the Fast Combat Support Ship (AOE-1 Class)? For purposes of simplification, the closed cycle gas turbine and reheat cycles have been ruled out. One may then consider the following five cycles: (Fig. 2.1)

1. Simple Cycle
2. Regenerative Cycle
3. Intercooled (external or internal)
4. Intercooled and regenerative
5. Any of the above in combination with a vapor cycle (e.g. steam) using the gas turbine exhaust

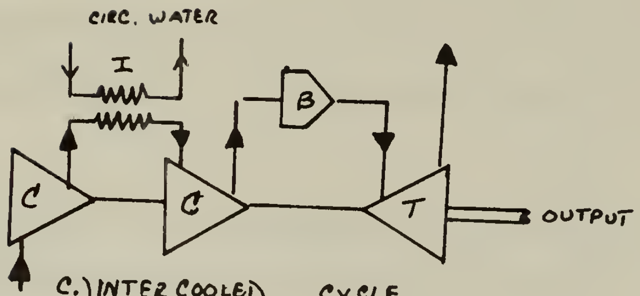
The simple cycle, while it is generally economically competitive with other prime movers, with respect to acquisition costs, has a rather low thermal efficiency and is dependent on low fuel costs unless the pressure ratio and turbine inlet temperatures are high. The regenerative cycle increases the thermal efficiency significantly at low pressure ratio, but the large acquisition costs coupled with size and weight considerations has ruled out its use by itself in many power applications. Additionally, the use of a regenerator alone reduces the output because of the additional pressure drops. The regenerator tends to have the most pronounced effect on conserving thermal energy at lower pressure ratios (less than about 5:1)



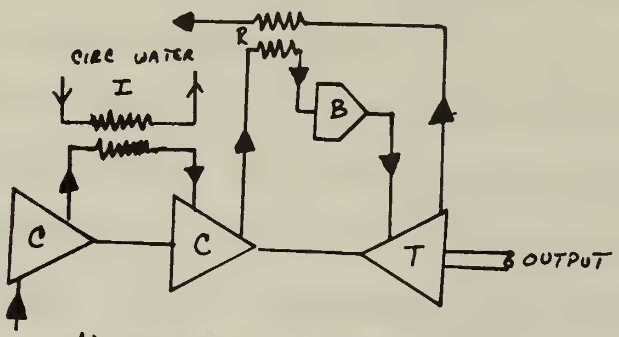
a.) SIMPLE CYCLE



b.) REGENERATIVE CYCLE



C.) INTERCOOLED CYCLE



d.) INTERCOOLED REGENERATIVE CYCLE

C, COMPRESSOR
T, TURBINE
B, BURNER (COMBUSTOR)

R, REGENERATOR
I, INTERCOOLER

Figure 2.1 Gas Turbine Cycle Combinations

depending to some extent on firing temperature. With most modern industrial gas turbines operating at higher pressure ratios, the object then becomes to reduce compressor horsepower in any available way. This calls for intercooling of the compressor between stages, either internally or externally. Here also, there is an associated pressure loss, but the reduction in air temperature with subsequent reduction in compressor work offsets this pressure loss, assuming a well designed heat exchanger.

The intercooled and regenerative cycle allows for a very efficient combination, since intercooling alone does not warrant the added cost. With a lower discharge temperature from the compressor due to intercooling, the energy in the exhaust may now be regained by using a regenerator, assuming the gas turbine exhaust temperature does not approach too closely the compressor outlet temperature. It would appear then that this cycle, from the standpoint of specific power and thermal efficiency, is the most desirable one.

An alternative way to improve the efficiency of the gas turbine cycle is to add a vapor cycle (generally steam) by using the waste gases from the gas turbine in a Rankine cycle. This method has been used in many different applications throughout the world where fuel costs are high. While a waste heat boiler could be used in concert with a regenerator, most applications would dictate one or the other insofar as insufficient heat would remain after the regenerator to generate any appreciable amount of steam in the heat recovery steam generator. Thus the consideration of a combined cycle for purposes of analysis will be made with an intercooler and a waste heat boiler. Comparative efficiencies of combined cycles are shown in Figure 2.2 (17). The choice of steam as the working fluid in the Rankine cycle

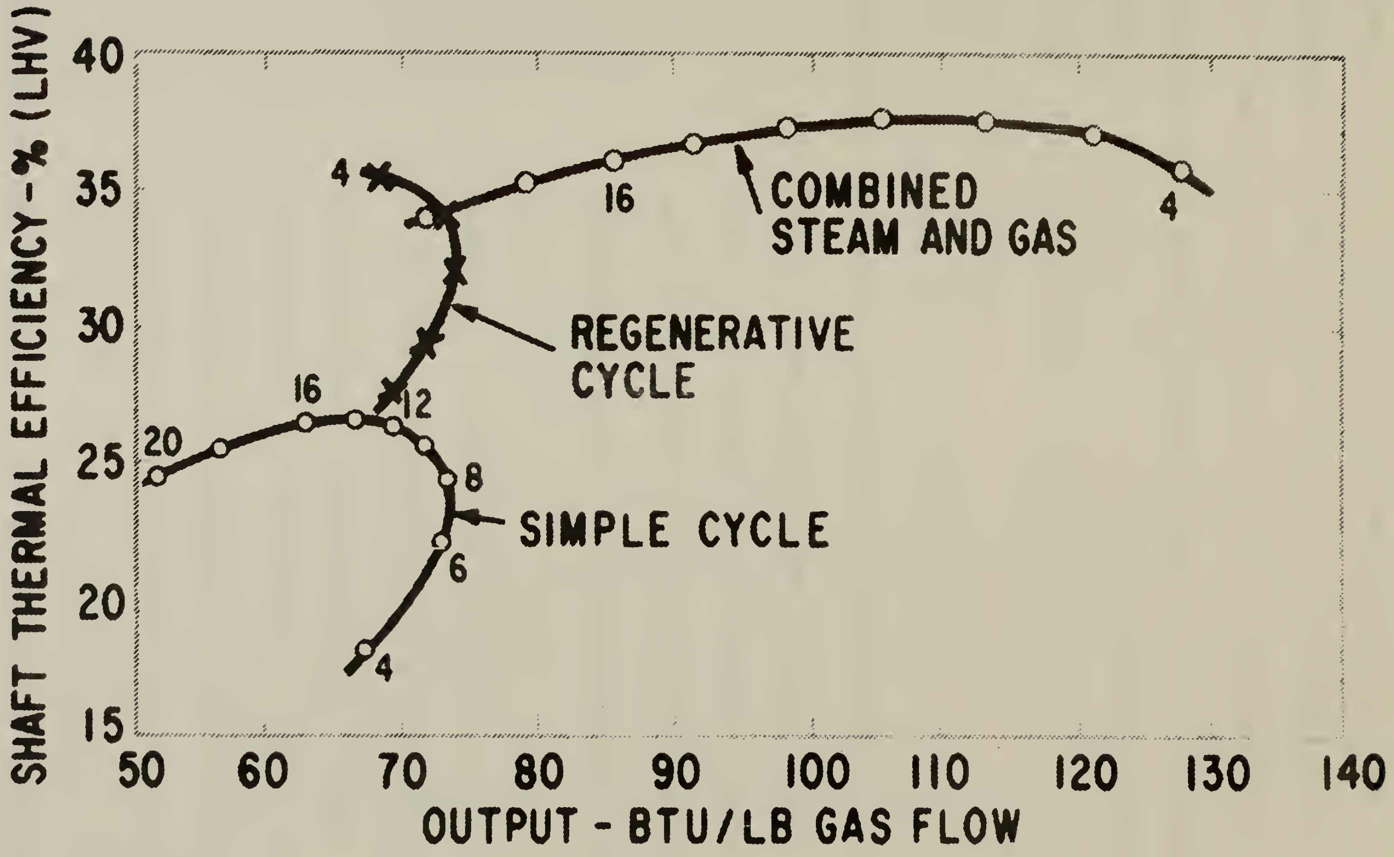


Figure 2.2 Comparative Efficiencies of Combined Cycles (17)

(Numbers along curve signify pressure ratio)

is an obvious one, as the steam technology (particularly shipboard) has been developed to a level that allows for reliability and ease of maintenance. Additionally, steam is desirable due to its properties which enable one to transfer the waste heat to another fluid which is capable of producing work down to a lower temperature level. Thus the combined cycle allows significant increase in thermal efficiency and attendant reduction in specific fuel consumption while not detracting too much from the basic simplicity of the gas turbine.

The combined cycle may take many forms. However, two large distinctions may be made in their use, particularly with respect to ships:

1. The heat recovery steam generator may be unfired or fired. The latter can be done in a variety of ways but generally a grid type fuel oil burner, spray nozzles, or a hot gas generator at boiler inlet are used;
2. The steam may be used to drive a steam turbine coupled to the propeller shaft to provide primary power; this is usually called COSAG with the gas turbine providing the "boost" power; the steam turbine may be used for secondary or "boost" power, with the gas turbine providing main power--this is usually referred to as COGAS or STAG. (18, 8)

Both of these systems have seen use in the propulsion of ships--COSAG plants have been used in European navies and the General Electric Company is about to install a steam and gas turbine (STAG) cycle on two roll-on, roll-off containerships for Broken Hill Proprietary Company, Ltd., in

Australia and in one ethylene/methane tanker for Hilmar Reksten in Norway. (See Figure 2.3)(19.) The John Sergeant did have a waste heat boiler in use for hotel loads, but the steam turbine was only coupled to the shaft in the event of an emergency.

A common reaction to the combined cycle is that it is very complex--the worst of both worlds, if you will. To some extent this is true--the complexity has gone up, the ability to make full power out to the shaft is dependent on more components--but these comments deserve a second look. First of all is the consideration of fuel consumption. While the simple cycle looks like a well packaged prime mover, it suffers from unacceptable specific fuel consumption and as Figure 2.4 (20) shows, 73% of the input heat energy is lost up the stack. (20) While the use of it as a prime mover for a naval vessel may reduce this argument somewhat, perhaps the Navy, in the corporate sense, should look more closely at life cycle costs with respect to fuel. For one ship the added fuel costs may be only a small portion of the operating costs, but for a class of 20 ships this can add up to a sizable portion of the operational budget. In the combined cycle, the waste heat boiler in the unfired configuration allows the steam evaporation rate to float and no sophisticated control system is required. Also the steam part of the combined plant is smaller than the equivalent steam plant yet produces the same power out to the shaft, with correspondingly smaller component and pipe sizes, etc.

An attractive variation of the combined gas and steam turbine uses the steam cycle to drive a supercharger "pre-compressor" rather than to couple a steam turbine directly to the shaft. The supercharged cycle would include an intercooler to use the heat sink of the sea, so readily

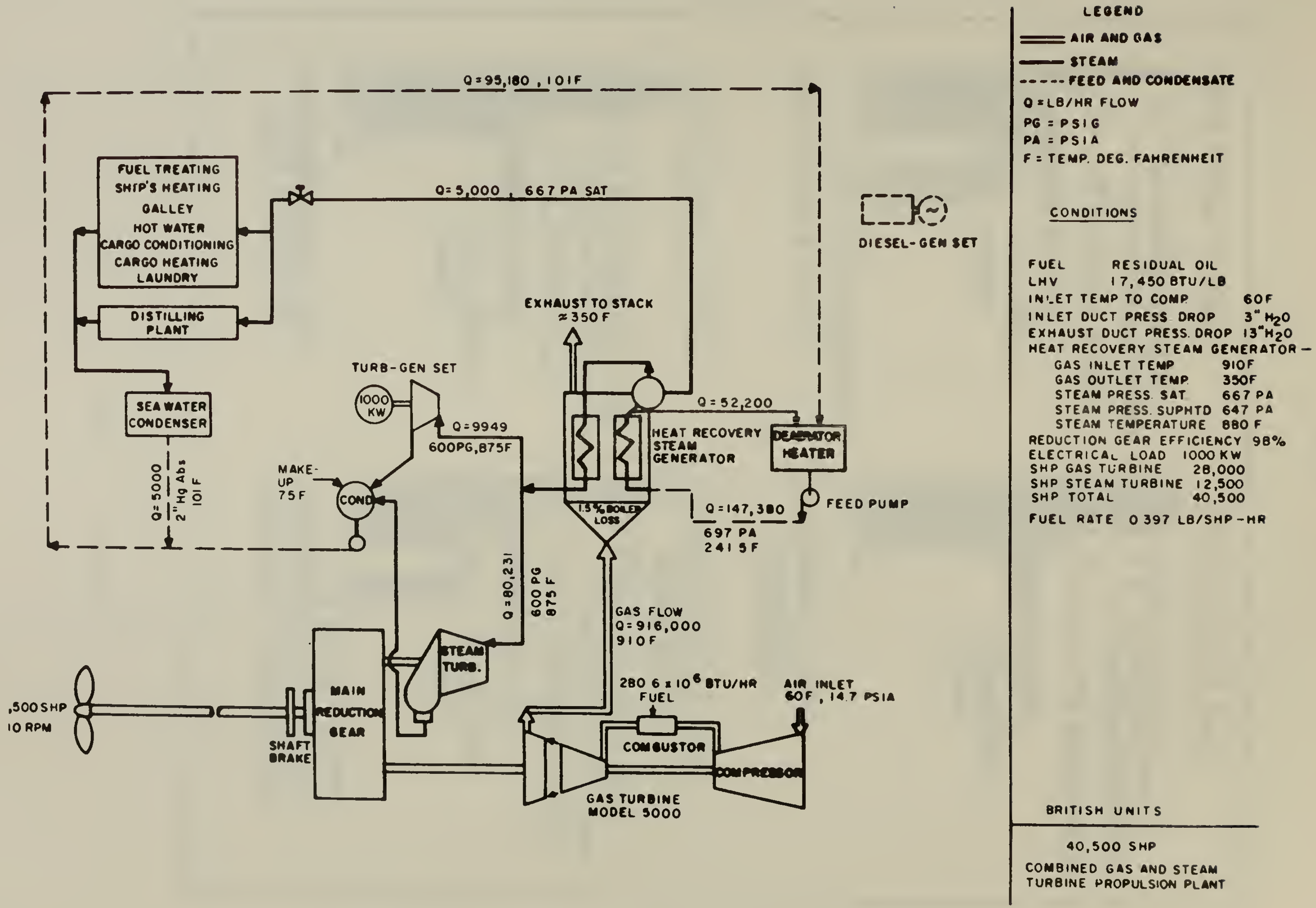


Figure 2.3 General Electric Company STAG Cycle (19)

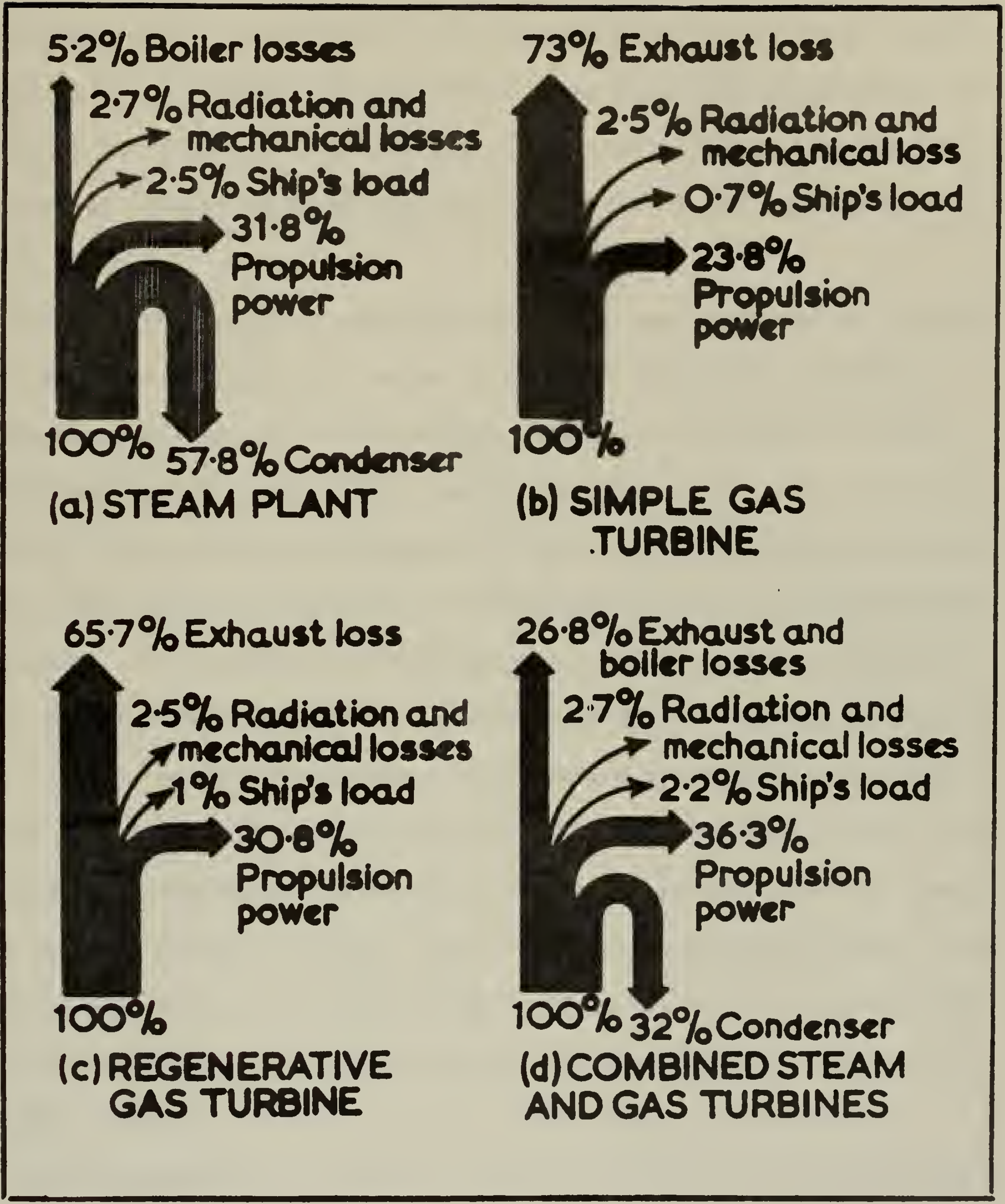


Figure 2.4 Energy Flow Diagram (20)

available on a ship. Shipboard gas turbine plants generally do not use intercooling and it would appear that more consideration of this prospect is warranted. In making a comparison with the steam plant, it should be noted that the STAG plant is normally not intercooled for shipboard use. This form of combined gas turbine and steam cycle is the subject of the thesis.

The supercharging of industrial heavy-duty gas turbines has not been practiced to any appreciable degree in the United States or anywhere in the world as well as can be determined. The basic principle is simple--the airflow through a gas turbine can be increased by the use of a supercharger to raise pressure at compressor inlet. The Lake Nasworthy combined cycle of West Texas Utilities was the first installation of a supercharged gas turbine. It consists of a Westinghouse W-301 gas turbine rated at 25,000 KW and supercharged at 53 in. water pressure (1.9 psig) by using a large motor driven fan. It used a Slough-Deflon evaporative intercooler and the resultant power out was 31,500 KW, a 26% increase in power. (21) The largest supercharged plants now in existence are part of a Dow Chemical Power Plant in Freeport, Texas. One of the three supercharged plants uses a Westinghouse W-501 gas turbine rated at 37,000 KW and the supercharged output is 43,000 KW, an increase of 16%. The supercharging fan for this plant also operates at 53 in. of water. (See Figure 2.5)(22,23.)

The thermodynamics of supercharging are developed from the basic analysis of one of the key gas turbine parameters, the flow function:

$$K_o = \frac{\dot{m}\sqrt{T_o}}{A_o P_o}$$

If the values of K_o, A_o and $\sqrt{T_o}$ are held constant and the inlet pressure is,

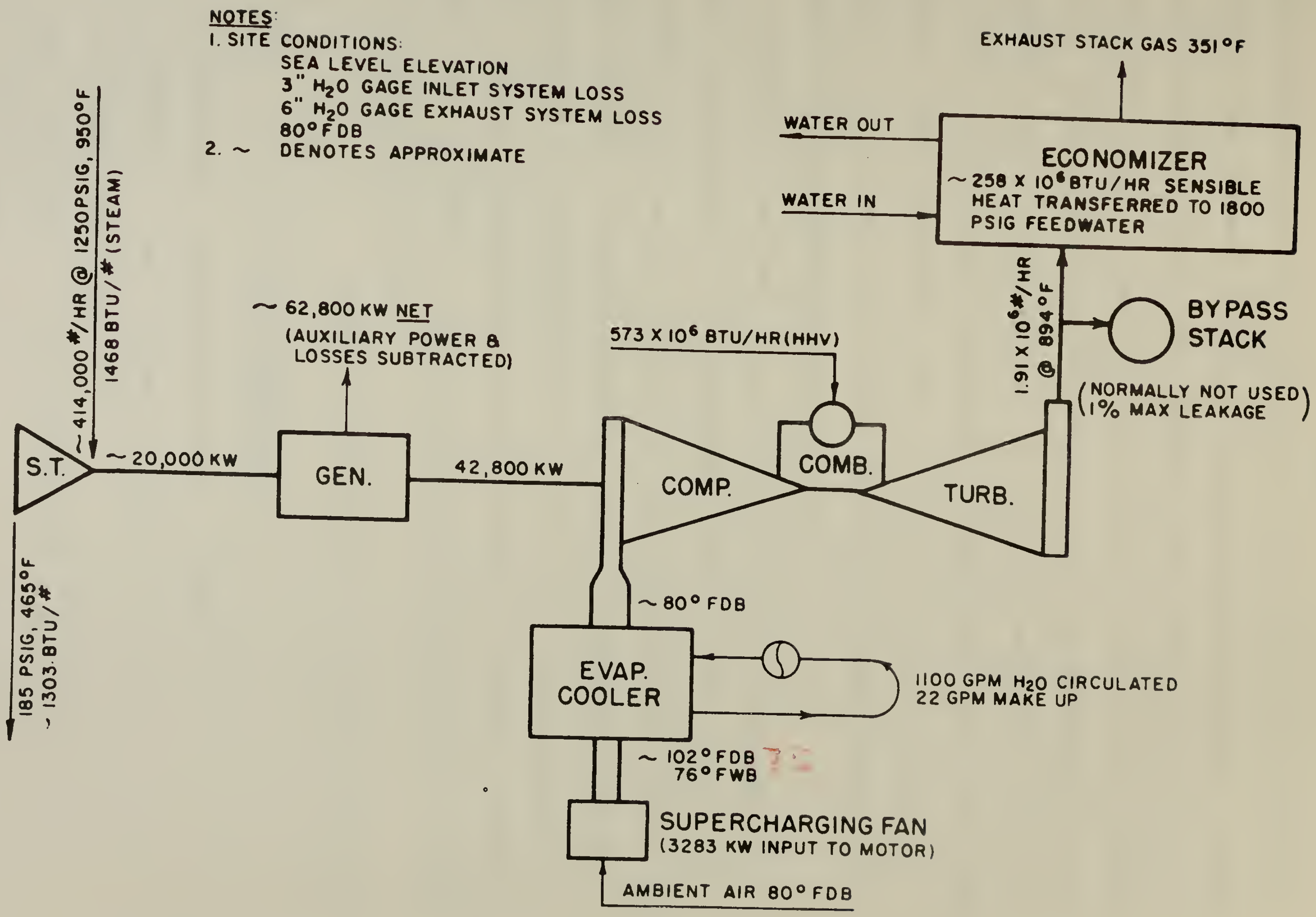


Figure 2.5 Cycle Performance Schematic, Westinghouse W-501 Supercharged Plant (22)

say doubled, the mass flow will double. Thus, if the inlet temperature stays the same the mass flow going through the machine doubles. The work of the turbine will not double, however, and the turbine exhaust temperature has now fallen. Noteworthy in the analysis is that the inlet temperature to the gas turbine must not change, and thus the intercooler is used to remove the heat of compression. In short, the gas turbine sees inlet air that remains at ambient temperature (ideally), but which is at a higher pressure. While there will be power required to drive the fan, the increase in power from the gas turbine will be two to three times that required for the supercharging compressor. (21)

One method of approach in calculating the effect of supercharging is given in reference (21). There the power output is first calculated by assuming the hypothetical case when the inlet pressure is the same as the exhaust pressure. The power determined from this hypothetical case is then adjusted for supercharging using a relationship between power increase and supercharge as shown in figure 2.6 (21). A linear relationship between percent increase in power resulting from supercharge will hold and can be expressed as follows:

$$\text{percent power increase} = K \times \text{percent supercharge}$$

where

$$K = 1 + \left(\frac{e \times {}^{\circ}R \times \text{lb/hr} \times g}{3.62 \times 10^{8}\ \text{HP}}\right)$$

e = supercharger compressor efficiency, percent

g = generator × gear efficiency, percent

$$\left.\begin{array}{l} {}^{\circ}R = \text{exhaust temperature} \\ \text{lb/hr} = \text{air flow} \\ \text{HP} = \text{power output} \end{array}\right\} \text{ at NEMA standard conditions}$$

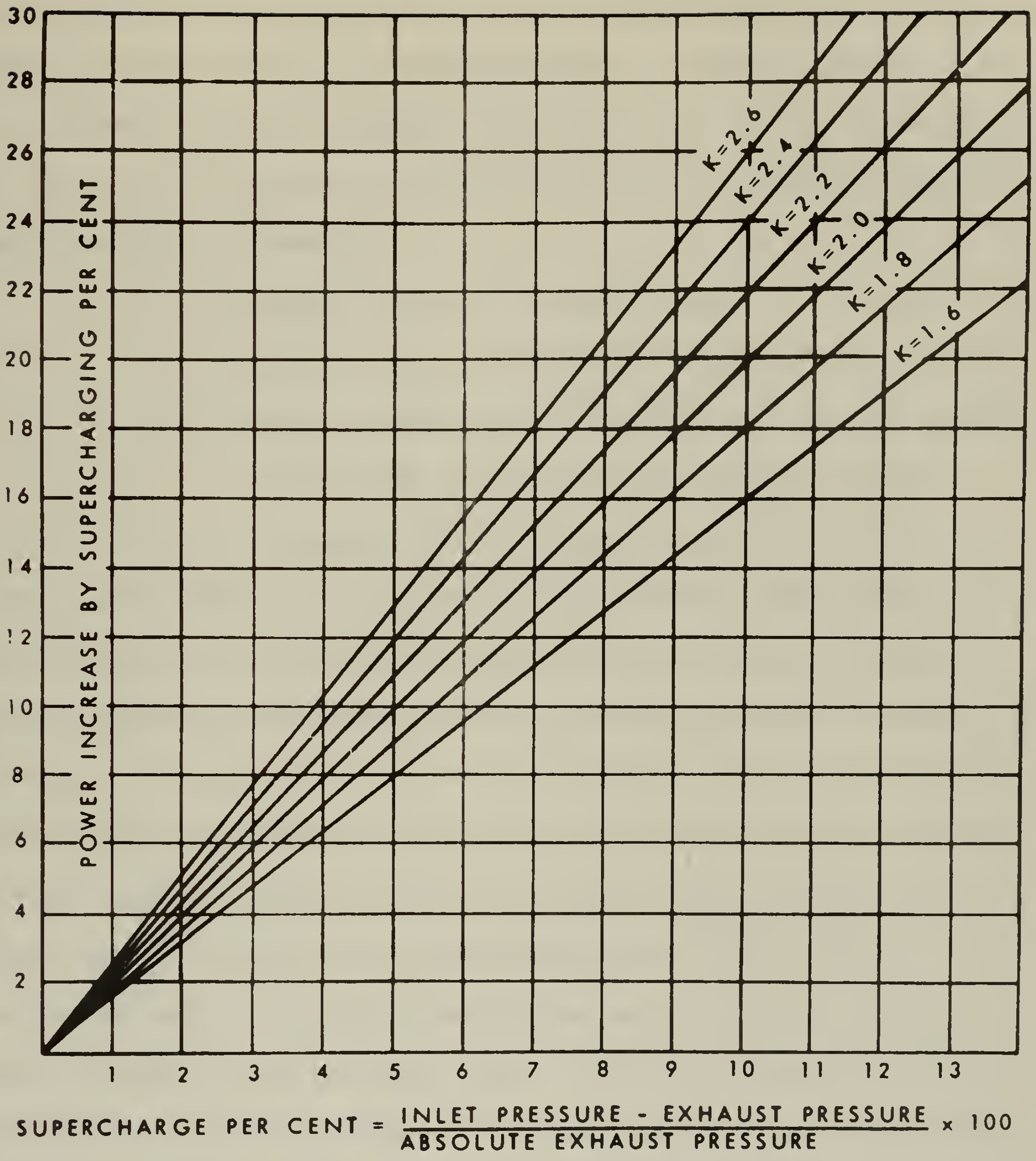


Figure 2.6 Gas Turbine Power Increase versus

Supercharging Pressure Ratio (21)

With the decision made to examine the feasibility of a supercharged combined cycle, the question which had to be answered was what would be a desirable pressure ratio for the supercharger? In order to analyze a propulsion system that could actually be used with "off the shelf" components (with little or no research and development costs) the pressure ratio was dependent on three parameters:

1. The shaft output of the AOE-1 Class, 2 × 50,000 HP;
2. The power ranges of available industrial gas turbines;
3. The power requirements for the steam plant and specifically the power required to drive the supercharger, either electrically or with steam.

With the shaft output fixed at 50,000 shaft horsepower (one shaft), a review of available heavy duty gas turbines indicated that the General Electric Company of Schenectady, New York, provided a possible turbine. Additionally, the company was involved in developing a STAG cycle for ship propulsion and some machines had "salt air" experience. While a gas turbine could have been chosen that had the supercharger directly coupled to the main compressor shaft, and an intercooler added, this would have required development of a specific gas turbine and would have somewhat reduced the flexibility of the power ranges. Thus the decision was made to go with a steam driven supercharger (to keep electric plant of the ship in the same configuration) external to the gas turbine, a tube and shell, platefin, or finned tube intercooler heat exchanger using seawater as a cooling medium, and an industrial gas turbine which would fit the power ranges. Two gas turbine of the Model Series 5000 were chosen, both two shaft machines, with one shaft coupled directly to the compressor and a

second or load shaft coupled directly to a controllable pitch propeller via variable angle inlet nozzles. The brief characteristics of the two turbines are:

	Fuel	BHP	THERMAL EFF, LHV	PRESS. RATIO	AIR FLOW	EXHAUST TEMP.
M5222 (B)*	Dist. oil	21,910	25.3	8.2	240.1 lb/sec	793°F.
M5322 (B)	Dist. oil	31,700	27.0	8.2	256.3 lb/sec	932°F.

*The Model Series 5000 notation uses the middle two digits as the basic horsepower rating, while the last digit indicates the number of shafts. There are both A and B machines, they differ from the 'straight' MS 5000 primarily insofar as the pressure ratio is concerned. The base machine has a pressure ratio of 7.6, while the A machine is lower at 6.9. The B machine has more compressor stages and the turbine nozzles and blades are at a slightly different angle from design A.

The analysis is shown in Table 2.1 and 2.2. To make a first cut for a feasible cycle, the following assumptions were made in these following calculations:

1. Supercharger polytropic efficiency, 0.86
2. Pressure loss in intercooler, 0.6 psia
3. Initially assumed that an additional stage would be added to the load turbine; polytropic efficiency, 0.86
4. Exhaust temperature set at 350° F.
5. Lower heating value of No. 2 distillate, 18,400

The results of these tables are plotted graphically in figures 2.7 and 2.8. Of prime importance now became the steam pressure and flow rate of

TABLE 2.1 Supercharging Pressure Level Feasibility G.E. MS 5222 (B)

Basic HP = 21,910, m = 240.1 lb/sec, exhaust 793° F.

		SUPERCHARGER PRESSURE RATIO			
		1.2	1.4	1.6	1.8
Supercharger Compress	$\gamma = 1.4$				
	Let $\eta_{poly} = 0.86$, $n = \frac{\gamma-1}{\eta_p \gamma} = \frac{.4}{.86 \times 1.4} = .3322$				
(1)	$\frac{\Delta T}{T_1} = 1-(PR)^n = 1-(PR)^{.3322}$	.0624	.1183	.1689	.2156
(2)	$\Delta T = 530° R \times (1)$	33.072	62.699	89.517	114.268
(3)	Pressure into G.T. = PR −.041 (ΔP in intercooler & piping)	1.159	1.359	1.559	1.759
(4)	HP of $G.T._{base} \times (3)$	25,393	29,776	34,158	38,539
(5)	$\dot{m}_{new} = \dot{m}_{base} \times (3)$	278.28	326.30	374.32	422.34
Additional Stage					
	$\gamma = 1.36$, $C_p = 0.26$, $\eta_{pol} = 0.80$, $n = \eta_p \left(\frac{\gamma-1}{\gamma}\right) = \frac{.80(.36)}{1.36} = .2118$				
(6)	$\frac{T_1}{T_2} = (PR)^n = (PR)^{.2118}$	1.0317	1.0671	1.0986	1.1270
(7)	$T_1 = 793 + 460 = 1253$				
(8)	$T_2 = T_1/(6)° R$	1214.5	1174.2	1140.5	1111.8
(9)	T_2, ° F (exhaust temperature)	754.5	714.2	680.5	651.8
(10)	ΔT, $793 - T_2$ ° F	38.5	78.8	112.5	141.2
(11)	Addt'l. HP, $\dot{m} c_p \Delta T = .3678 \dot{m} \Delta T$	3,940	9,457	15,488	21,934
(12)	Total HP to shaft	29,333	39,233	49,646	60,473

TABLE 2.2 Supercharging Pressure Level Feasibility G.E. MS 5322 (B)

Basic HP = 31,700, $\dot{m}$ = 256.3 lb/sec, exhaust = 932° F.

$\gamma = 1.4$

SUPERCHARGER PRESSURE RATIO:

	$\eta_p = 0.86$, $n = .3322$	1.2	1.4	1.6	1.8
(1)	$\frac{\Delta T}{T_1} = 1 - (PR)^n = 1 - (PR)^{.3322}$	.0624	.1183	.1689	.2156
(2)	$\Delta T = 530° R \times (1)$	33.072	62.699	89.517	114.268
(3)	Pressure into G.T. = PR -.041	1.159	1.359	1.559	1.759
(4)	HP of $GT_{base} \times (3)$	36,740	43,080	49,420	55,760
(5)	$\dot{m}_{new} = \dot{m}_{base} \times (3)$	297.05	348.31	399.57	450.83

Additional Stage

$\gamma = 1.36$, $c_p = 0.26$, $\eta_p = 0.80$ $n = \frac{(\gamma - 1)\eta_p}{\gamma} = .2118$

(6)	$\frac{T}{T_2} = (PR)^n = (PR)^{.2118}$	1.0317	1.0671	1.0986	1.1270
(7)	$T_1 = 932 + 460 = 1390° R$				
(8)	$T_2 = \frac{T_1}{(6)}$ ° R	1349.2	1304.4	1270.0	1235.1
(9)	T_2° F (exhaust temperature)	889.2	844.4	810.0	775.1
(10)	ΔT, 932 - T_2° F	42.8	87.6	122.0	156.9
(11)	Addt'l. HP, $\dot{m}c_p\Delta T$	4,676	11,222	17,929	26,300
(12)	Total HP to shaft	41,416	54,302	67,349	82,060

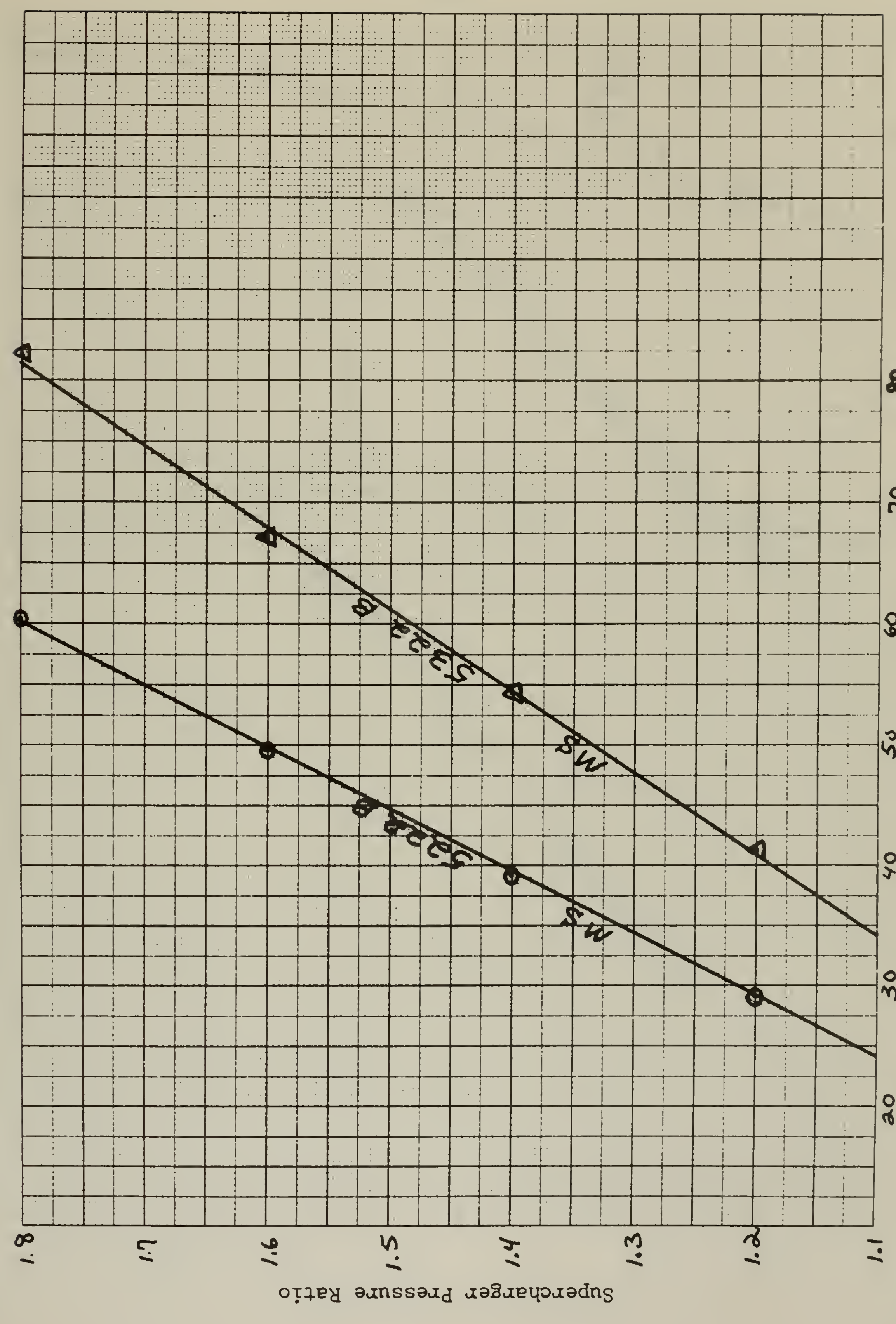


Figure 2.7 Shaft Horsepower--Effect of Supercharging MS 5222 (B) and 5322 (B)

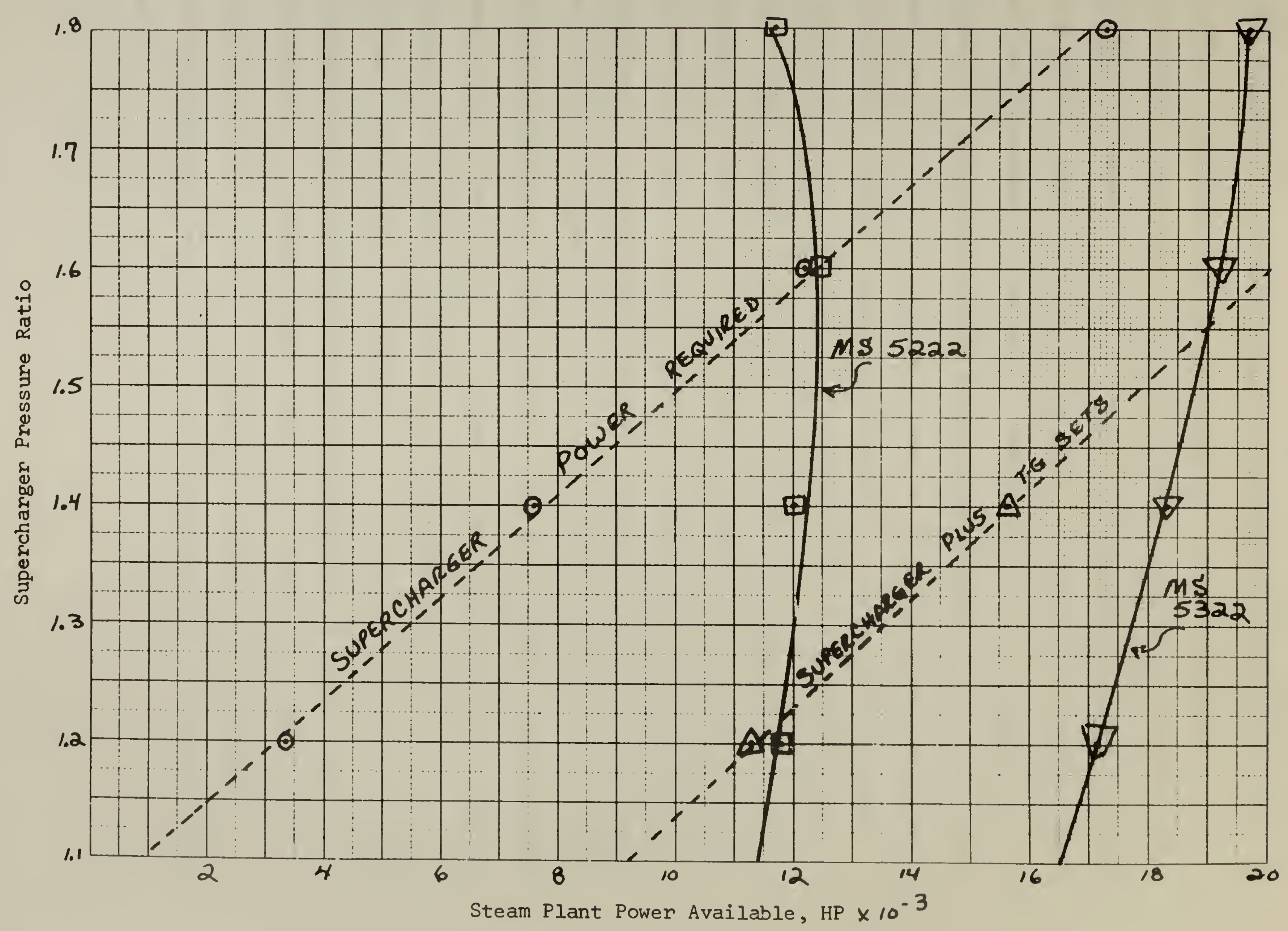


Figure 2.8 Steam Cycle Power Output

steam necessary to at least drive the supercharger and to maintain the electrical load, while having a pressure ratio that would give the required shaft horsepower. As may be seen from Figure 2.7 the 5222B machine indicated a pressure level of 1.6 for the supercharger to be the most desirable one, while the pressure ratio for the 5322 machine produced the required horsepower at just over a pressure ratio of 1.3. Looking at Figure 2.8, one notes the basic supercharger operating line, this is the power required to drive the supercharger at the various pressure ratios. Additionally, however, there is a requirement to drive 4-1500KW turbo-generators as well and this line is plotted. The available steam flow for the heat recovery steam generator for each gas turbine with a stack temperature of 350° F. is then plotted in order to see where the desirable pressure ratio lies. At first glance, it appears that the 5222 machine operating with a supercharged pressure ratio of 1.2 is the most desirable, but referring to Figure 2.7, it can be seen that the power levels at this pressure ratio produce only 30,000 SHP. The other machine intersected at over a pressure ratio of 1.5--this gave much more power than was needed. Therefore, the decision was made to use a pressure ratio of 1.4 and the 5322 gas turbine--this gave sufficient power to drive the supercharger and turbo-generator sets and provided sufficient power out to the shaft to drive the ship at maximum speed. Admittedly, there is more power than needed, but for a 'first cut' feasibility and without consideration of gear efficiency and other losses, this decision appears to have been satisfactory.

3. THE SUPERCHARGED GAS TURBINE AND STEAM CYCLE

3.1 General Description

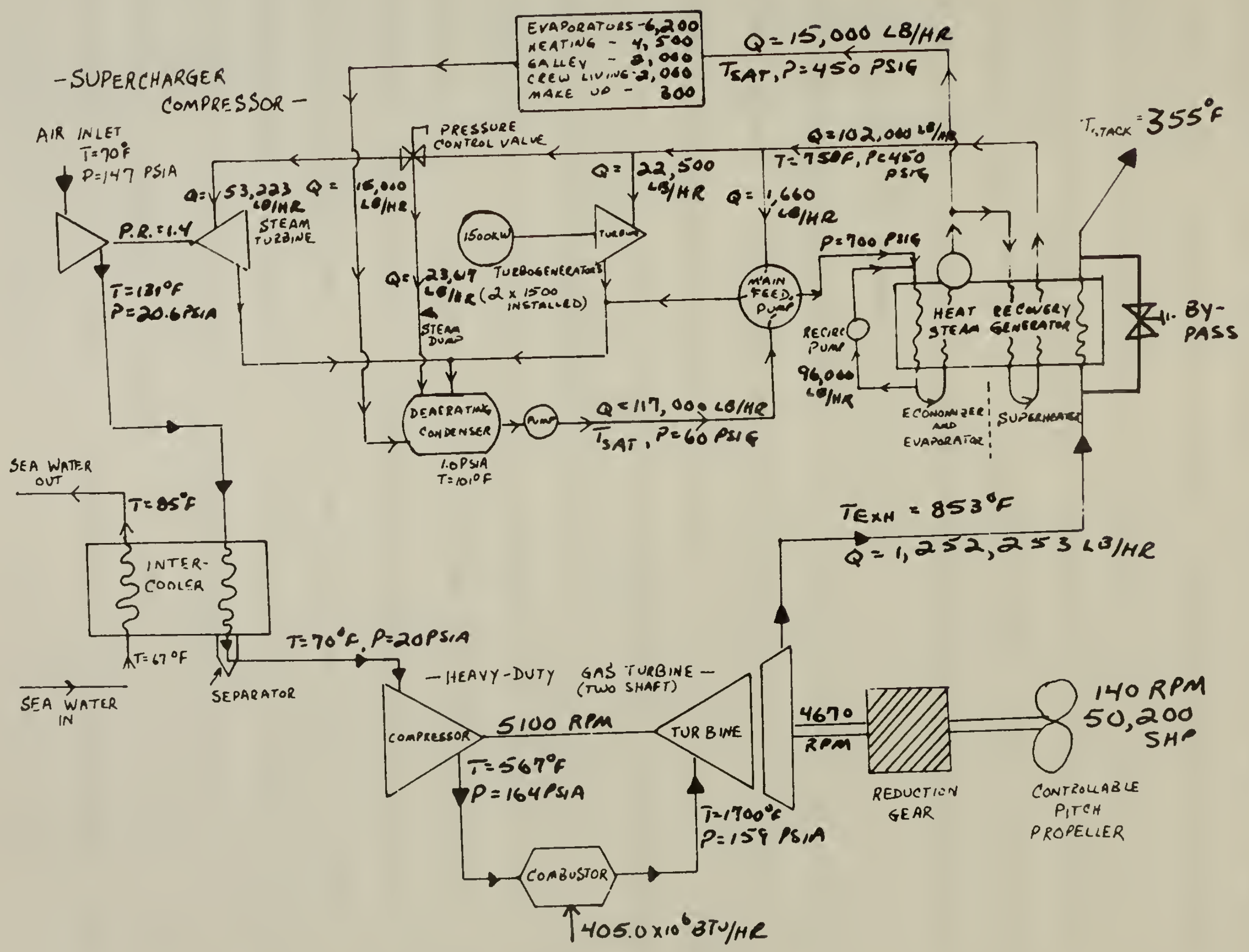
The basic plant setup at full power for this cycle is shown in Figure 3.1. Table 3.1 shows the critical parameters in the cycle at full power, 17-knot endurance and 12-know replenishment powers. The off-design points will be addressed in more detail in section 3.4, and therefore this section describes the plant in the full power condition, unless otherwise noted.

Looking at Figure 3.1, air is taken at ambient conditions at 70° F. and 14.7 psia into the supercharger compressor. The inlet section contains a standard inlet and filter house, de-mister, and de-icer--all primarily to keep foreign particles and sea salts out of the system. The air is pressurized to 20.6 psia, then passed through an extended finned tube intercooler, with sea water design inlet temperature at 67° F. and the sea water running through the tubes, with the gases passing around them in a conventional crossflow unmixed heat exchanger design. The purpose of the intercooler is to reduce the heat of supercharging by use of the ocean, a large heat sink readily available. From here the air passes to the inlet of the heavy-duty gas turbine at a temperature of 70° F. and a pressure of 20.0 psia. The pressure loss suffered at inlet to the main compressor is due to the intercooler (.349 psia) and the ducting, separator, and compressor inlet section (.251 psia).

The General Electric MS 5322 gas turbine consists basically of a compressor section, a combustion section and a two-stage expansion section. (24) In the supercharged cycle, the temperature from the compressor is

TABLE 3.1 SUPERCHARGED CYCLE CHARACTERISTICS

		FULL POWER	17-KNOT ENDURANCE	12-KNOT REPLENISHMENT
1.	Air Inlet Temp., ° F.	70.0	70.0	70.0
2.	Air Inlet Pressure, psia	14.7	14.7	14.7
3.	Supercharger Pressure Ratio	1.4	1.17	1.13
4.	Supercharger Discharge Temp., °F.	131.0	100.0	85.0
5.	Supercharger Discharge Pressure, psia	20.6	17.2	16.6
6.	Sea Water Inlet Temperature, Intercooler, ° F.	67.0	67.0	67.0
7.	Sea Water Outlet Temperature, Intercooler, ° F.	85.0	83.0	74.0
8.	Gas Turbine Inlet Temp., ° F.	70.0	70.0	70.0
9.	Gas Turbine Inlet Pressure, psia	20.0	16.6	16.0
10.	G.T. Compressor Dishg. Temp., °F.	567.0	430.0	430.0
11.	G.T. Compressor Dishg. Pressure, psia	164.0	84.5	81.3
12.	Combustor Heat Rate, Btu/hr	405×10^6	109×10^6	70×10^6
13.	G.T. Firing Temperature, °F.	1700.0	1230.0	1110.0
14.	G.T. Turbine Inlet Pressure, psia	159.0	82.0	78.7
15.	G.T. High Pressure Speed, RPM	5100	4060	4060
16.	G.T. Low Pressure Speed, RPM	4670	2800	2000
17.	Shaft Horsepower, Ship	50,200	11,100	6,750
18.	Shaft Speed, RPM	140	84	60
19.	Propeller Pitch, Percent	100	100	100
20.	Second-Stage Variable Angle Nozzles, Degrees	0	+2	+4
21.	G.T. Exhaust Temperature, ° F.	853.0	703.5	624.5
22.	G.T. Exhaust Flow, lb/hr	1,252,253	856,000	826,000
23.	HRSG Stack Temperature, ° F.	355.0	375.0	400.0
24.	HRSG Recirculation Rate, lb/hr	96,000	43,000	27,000
25.	HRSG Back Pressure, "H_2O	9	4.4	4.1
26.	Superheated Steam Flow, lb/hr	102,000	39,650	24,930
27.	Saturated Steam Flow, lb/hr	15,000	15,000	10,500
28.	Superheater Steam Temp., °F.	750.0	680.0	620.0
29.	Superheated Steam Pressure, psi	450.0	450.0	450.0
30.	HRSG Steam Drum Pressure, psi	475.0	475.0	475.0
31.	Gas Temp. Leaving Superheater, ° F.	800.0	680.0	615.0
32.	Gas Temp., Leaving Evaporator	510	480	475
33.	Turbogenerator Load, KW	2000	570	1115
34.	Turbogenerator Steam Flow lb/hr	22,500	7,550	13,750
35.	Diesel Generator Load, KW	0	0	1000
36.	Supercharger Steam Flow, lb/hr	53,223	16,257	9,804
37.	Steam Dumped, lb/hr	23,617	14,984	796
38.	Feed Pump Steam Flow, lb/hr	1,660	859	580
39.	Feed Pump Dischg. Pressure, psig	700	700	700



SUPERCHARGED GAS TURBINE AND STEAM PLANT

- FULL POWER -

LEGEND

AOE-1 Full Power Cond.:
Steam Plant
2 x 50,000 SHP
140 RPM
Design fuel rate .632
4000 KW Electric Ld.

SUPERCHARGED PLANT AT FULL POWER:

2 x 50,200 SHP
140 RPM
Supercharger SHP 7750
4000 KW Electric Ld.
(4 TG sets running)
Combined cycle overall thermal efficiency: 0.316
All purpose design fuel rate: 0.444 $\frac{\text{lb}}{\text{SHP-hr}}$
Navy Distillate fuel
LHV: 18,100
Fuel/air ratio:1.0185

Figure 3.1 Supercharged Power Plant Setup--Full Power

567° F. at a pressure of 164 psia enroute to the combustor, which burns Navy Distillate fuel. There is a 3% pressure loss in the combustion cans which deliver the gases to the high pressure turbine at a firing temperature of 1700° F. Polytropic (stage) efficiency of the HP turbine is 0.87. The gases then pass to the second stage or low pressure turbine (load turbine) via variable angle nozzles, which control residual power basically by changing the velocity triangle of the load turbine. The expansion in the LP turbine is accomplished in one stage ($\eta_p = .84$) instead of an added stage as assumed in feasibility study in Chapter Two. Leaving losses remain at an acceptable level. The load turbine is coupled directly to the main reduction gear and via a 33.4:1 reduction, to the controllable and reversible propeller.

The exhaust gas temperature is 850° F. and the gas flow is 341.5 pounds per second. These gases are ducted directly to an unfired heat recovery steam generator which contains economizer, evaporator, and superheater sections. The economizer is fitted with a recirculation pump to bring the feedwater temperature up to 240° F., and the heat recovery steam generator is also fitted with a by-pass to the uptakes (stack) if the steam cycle is not in use due to required maintenance underway. The steam generator operates at a steam drum pressure of 475 psig and line pressures and temperatures to operate machinery as shown in Figure 3.1 are 450 psig, 750° F. Some of the steam generated is taken directly from the steam drum (15,000 lbs/hr) and passed through reducers for use in the evaporators, for heating and galley steam, and for crew living. The superheated steam is used to drive the steam driven supercharging compressor, two 1500 KW turbogenerators, and the main feed pump. The

turbogenerators and feed pump operate at a constant inlet pressure of 450 psig--line pressure is controlled by a pressure control valve located at the inlet to the supercharging steam turbine. This pressure control valve dumps any excess steam to the main condenser and admits steam to the supercharger turbine as the demand changes, as explained in section 3.4.1. All three major components exhaust to one large main deaerating condenser operating at 1.0 psia (2" Hg) and saturated steam temperature of 101° F. The condenser has the function of removing oxygen and dissolved gases from the feedwater, thus it is very similar to condensers used on submarines, which have no deaerating feed tank. Additionally, the condenser must be able to admit large quantities of "dumped" steam via breakdown orifices and nozzles.

Positive suction is taken from the hotwell of the main condenser by a condensate pump which discharges at approximately 60 psig to the suction side of the steam driven main feed pump. The feed pump discharges feedwater to the steam drum of the heat recovery steam generator at a pressure of 700 psig and temperature of 105° F. This low feed temperature coming from the feed pump is heated to 240° F. by an economizer recirculating pump in order to reduce corrosion in the economizer. Hydrazine or sodium sulfite will be added to the feedwater as oxygen scavengers before the feedwater enters the steam generator. It is envisioned that a continuous monitor for oxygen and chlorides would be installed on the discharge side of the feed or condensate pump.

3.2 Gas Turbine Propulsion System

3.2.1 General Description of the Gas Turbine (24)

A schematic diagram of the MS 5322 General Electric gas turbine is

shown in Figure 3.2. (24) The compressor section is a 16-stage compressor with a pressure ratio of 8.2:1. The compressor casing and blading (bending stresses) at supercharged power levels require no modification to present design. The gases flow to a combustion system composed of twelve 10-inch diameter chambers located around the outside of the compressor casing and enclosed in a combustion wrapper or shell. Here Navy distillate is burned at the rate of 405 million Btu/hr at full power at a firing temperature of 1700° F. and these hot gases pass directly to the high pressure turbine. The high pressure, or compressor turbine, rotates at 5100 RPM at full load. The first-stage nozzle has twelve segments with three vanes which enable trailing edge cooling by using compressor air. The gases then pass to the low pressure (load) turbine through 32 precision cast nozzle vanes mounted on shafts that extend radially through the turbine casing. Each shaft carries a lever arm that connects to the control ring which rotates around the casing to vary the nozzle throat area. The nozzle control ring is actuated by a hydraulic cylinder operated by the control system, which changes the angle setting depending on the load.

A detailed description of the gas turbine with photographs of the above components is contained in Appendix C.

3.2.2 Supercharger Compressor

As discussed in Chapter 2, the decision was made to operate the supercharger at a pressure ratio of 1.4 to get the desired output. This pressure ratio presented a problem in the design of the propulsion plant. This pressure ratio might be said to be located in the twilight zone of generally available characteristics, i.e., too high for a large fan

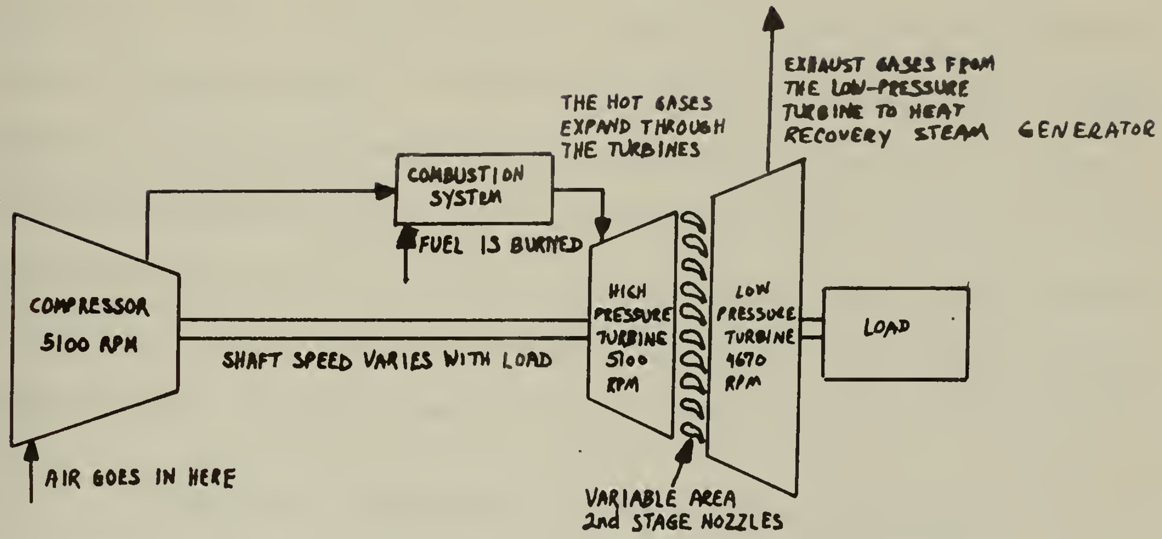

EXHAUST GASES FROM THE LOW-PRESSURE TURBINE TO HEAT RECOVERY STEAM GENERATOR
THE HOT GASES EXPAND THROUGH THE TURBINES
COMBUSTION SYSTEM
FUEL IS BURNED
COMPRESSOR 5100 RPM
HIGH PRESSURE TURBINE 5100 RPM
LOW PRESSURE TURBINE 4670 RPM
LOAD
SHAFT SPEED VARIES WITH LOAD
AIR GOES IN HERE
VARIABLE AREA 2nd STAGE NOZZLES

Figure 3.2 Schematic Diagram of the Two-Shaft Heavy Duty Marine Gas Turbine (24)

(50-60 in. of water) and yet too low for most of the many aircraft engine compressors which do not go below pressure ratios of 2.0:1. Finding compressor characteristics which would operate throughout the range of the cycle was necessary to insure a feasible design and prevent surge or choke at off-design points.

The compressor characteristics for the supercharger are shown in Figure 3.3. They were developed by taking M.I.T. Gas Turbine Laboratory curves (Figures 3.4 and 3.5) (25)(26) for a single-stage axial flow compressor and combining them into a two-stage axial compressor to obtain the desired characteristics. The calculation procedure for obtaining the characteristics, as well as sizing and speed determination may be found in Appendix A, Thermodynamic Considerations of the Cycle. Basic design features of the compressor may be found in Table 3.2.

TABLE 3.2 SUPERCHARGER COMPRESSOR CHARACTERISTICS

Type: Two-stage axial flow compressor
Prime Mover: 12-stage steam turbine, direct coupling
Pressure Ratio: 1.4:1
Stage Efficiency: 0.901
Speed: 4950 RPM at full rating
Diameter: 3.89 feet
Length: 3.00 feet (estimated)
Blade Speed (max): 1000 ft/sec
Blade Length: 1.93 feet
Blade Hub/Tip Ratio: 0.55
Flow Rate (max): 348.5 lbs/sec
Flow Area: 8.13 ft^2

3.2.3 Intercooler

The intercooler is of the extended surface circular finned tube type and has basic dimensions of 5 ft. by 5 ft. by 20 ft. The design for feasibility used procedures in reference (27) and the results are shown in Table 3.3. The decision to use circular-finned tubes instead of plate-fin

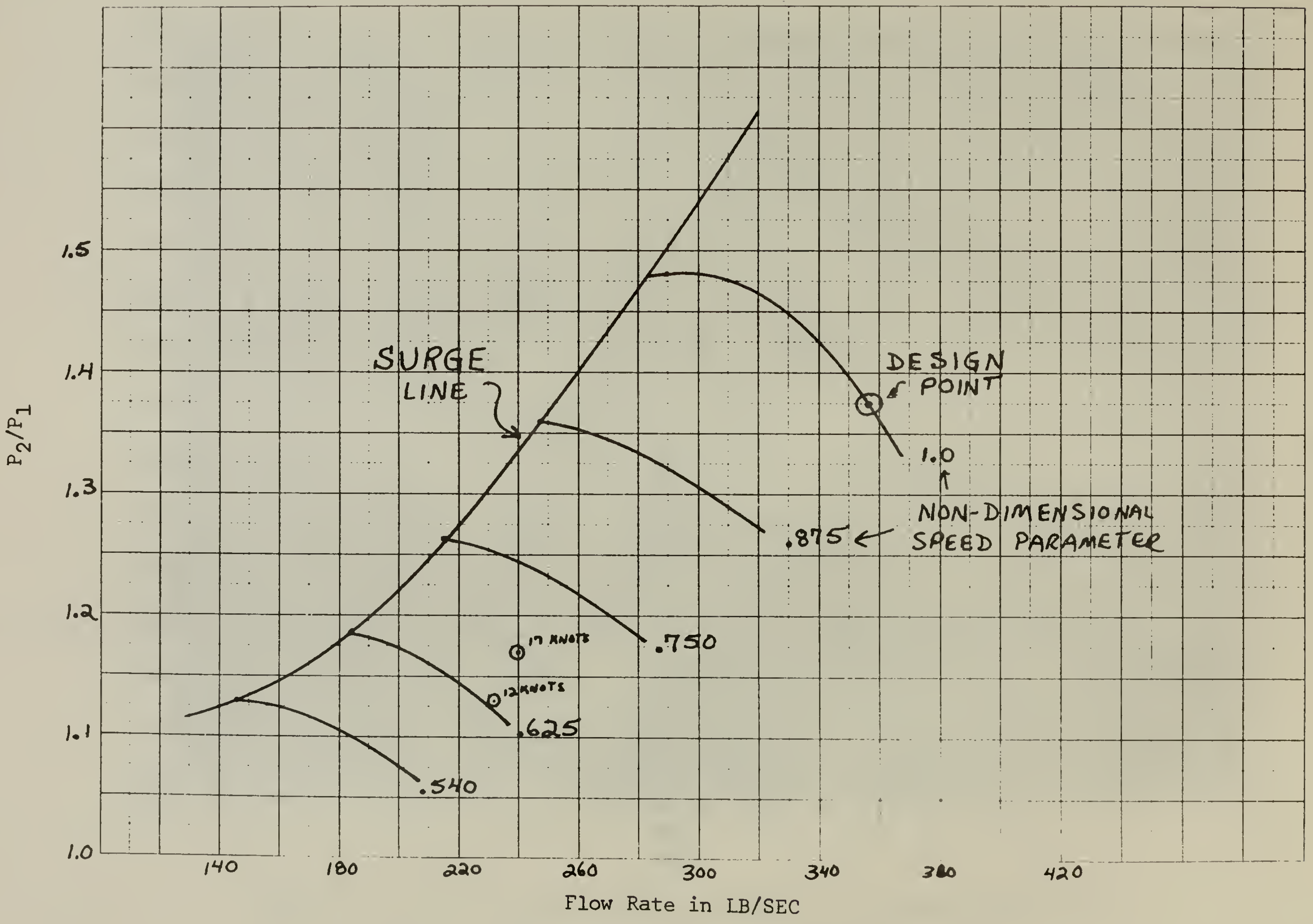


Figure 3.3 Supercharger Compressor Characteristics

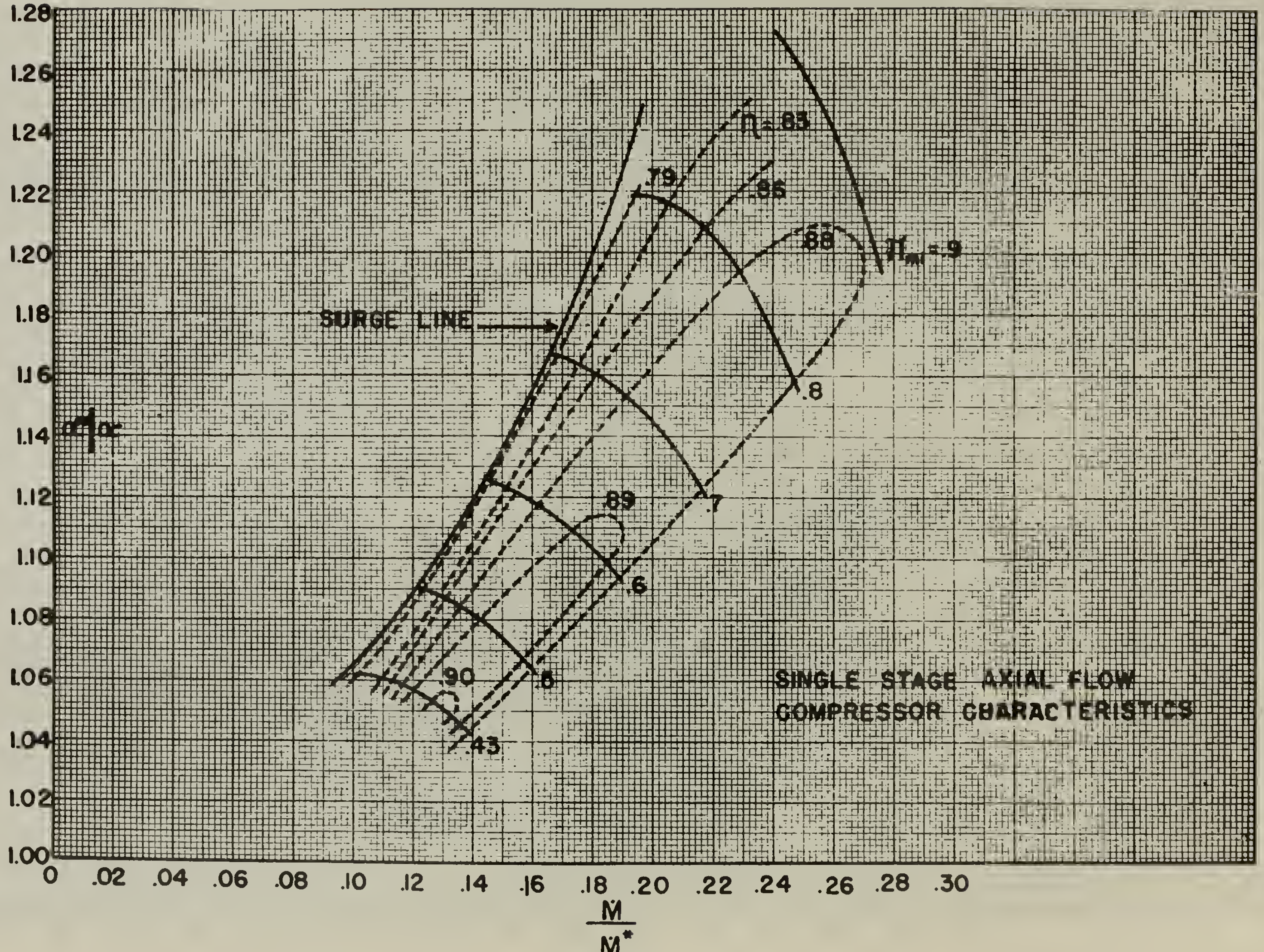


Figure 3.4 M.I.T. Gas Turbine Lab Curve No. 6, P_2/P_1 versus $\dot{m}/\dot{m}^*$ (25)

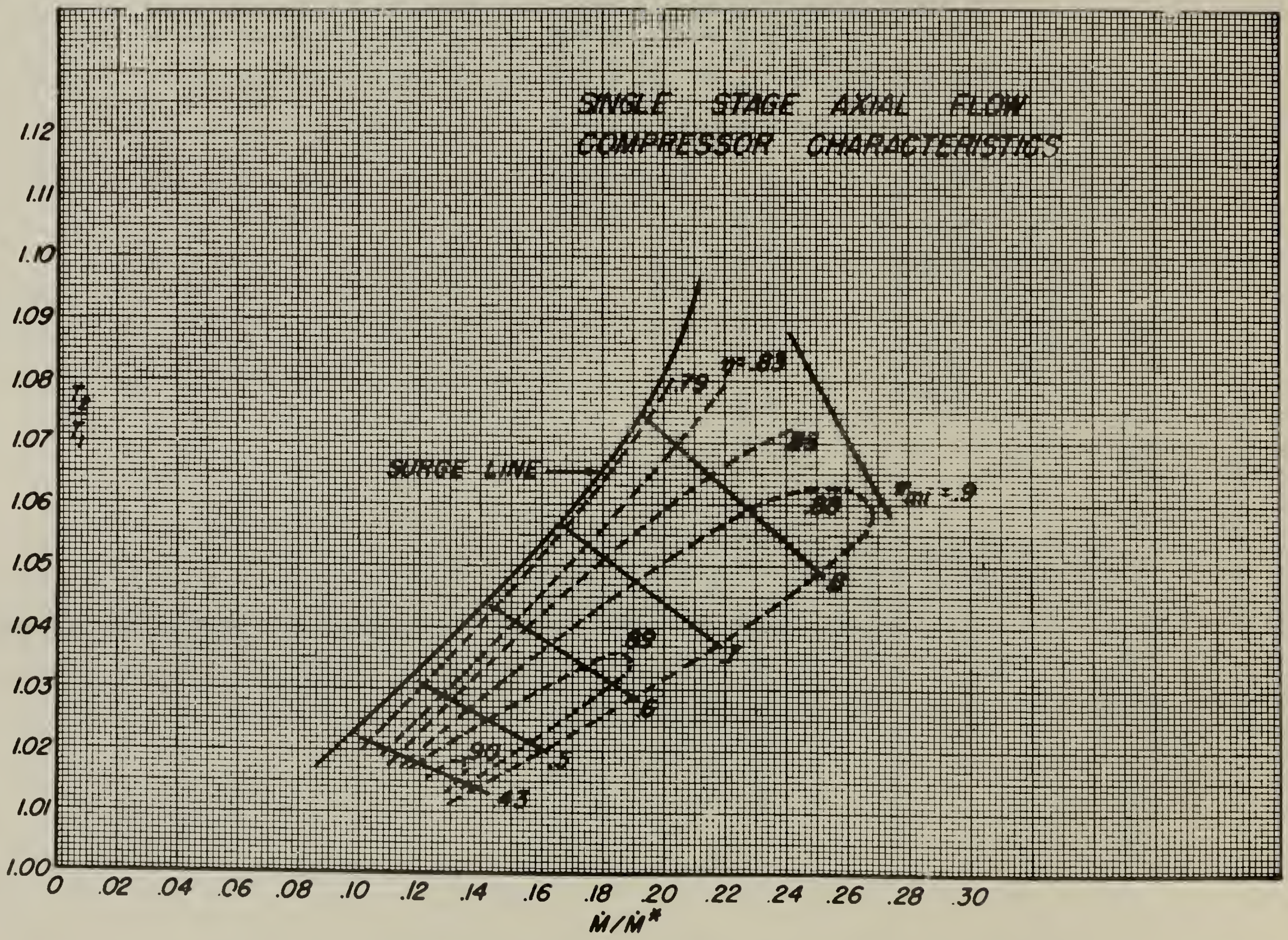


Figure 3.5 M.I.T. Gas Turbine Lab Curve No. 7, T_2/T_1 versus $\dot{m}/\dot{m}^*$ (26)

TABLE 3.3 INTERCOOLER CHARACTERISTICS (FULL POWER)

(See Appendix B.)

Type: Continuous finned circular tube (Jameson Data)
No. CF - 3/4 J-E (27) (See Fig. B.1) (68)

Air Flow Rate:	1,277,000 lb/hr
Air Temperature, Entering:	131° F.
Air Pressure, Entering:	20.6 psia
Air Temperature, Leaving:	70.2° F.
Air Pressure, Leaving:	20.251 psia
Salt Water Flow Rate:	3,000,000 lb/hr
Salt Water Temperature, Entering:	67° F.
Salt Water Temperature, Leaving:	85° F.
Fin Material and Conductivity:	Aluminum, K = 100 Btu/hr-ft ° F.
Fin Length:	0.2595 in.
Fin Thickness:	0.012 in.
No. of Tubes:	1321
Tube Diameter (I.D.):	0.774 in.
Overall Heat Transfer Coefficient:	21.0 Btu/hr-ft^2 ° F.
Number of Heat Transfer Units (NTU):	3.6
C_{min}/C_{max}:	0.105
Overall Surface Efficiency:	0.90
Intercooler Effectiveness, ε:	0.95
Gas Pressure Drop:	0.349 psia
Intercooler Length:	20 feet
Intercooler Width:	5 feet
Intercooler Height:	5 feet

or other design was predicated on the fact that salt water flowing in the tubes was used as a cooling medium. A plate-fin intercooler might allow undesirable salts into the air passages in the event of leakage, and this could cause complete failure of the hot gas section of the gas turbine.

Intercooling was used to enable the gas turbine to use the same inlet temperature as ambient but at a higher pressure level, thereby increasing power. While intercooling is generally more widely used at higher temperatures, its presence in the cycle does increase the overall thermal efficiency and uses the ocean to reject heat. Most gas turbine designs for marine propulsion do not use intercoolers, most probably due to size and weight constraints. However, in a large ship with relatively spacious engine rooms, an intercooler may be added quite readily. The intercooler was placed into the cycle as a separate component (versus integral with gas turbine compressor) to allow an existing gas turbine. Sea water will be taken from the main scoop injection section of the main condenser, with a salt water circulating pump (motor driven) to provide flow through the intercooler at low speeds.

One advantage of using an intercooler in a gas turbine cycle is the tendency for the intercooler to remove sea salts which pass through the inlet house and demister which are located upstream of the supercharger compressor. As may be seen from Figure 3.1, a separator has been added immediately after the intercooler. This is of the louvered inertial type, similar to the Dyna-Vane air cleaner cell. (See Figure 3.6)(28.) The cooling of the air in the intercooler at the higher pressures would tend to condense and settle out entrained water and sea salts. While an

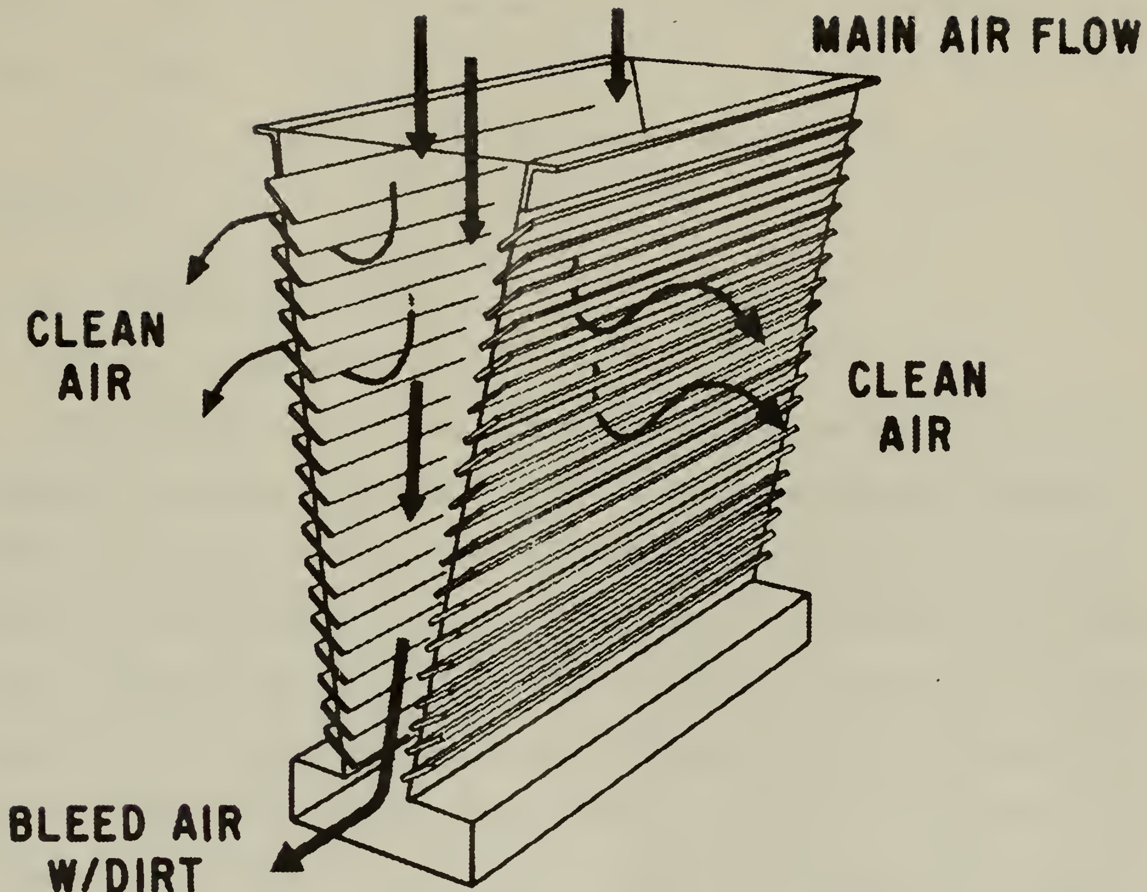


Figure 3.6 Louvered Inertial Separator (28)

[illegible]

additional separator adds to the cost of the plant and may be somewhat conservative, the results of compressor fouling can reduce the efficiency and power out from the gas turbine. (29) Additionally, as stated before, sodium in large quantities on hot gas parts can result in catastrophic turbine failures. Average sea-salt aerosol concentrations are shown in Figure 3.7 (30)

A detailed analysis of the intercooler design may be found in Appendix B.

3.2.4 Transmission System

3.2.4-a. Reduction Gear

The decision to use a standard reduction gear was predicated on the fact that the plant uses a controllable and reversible pitch propeller. Gear technology is such that this component is perhaps the most reliable element in the transmission system. The reduction gear chosen is a single-input, single-output, double helical dual torque path type with a separate segmented main thrust bearing as shown in Figure 3.8 (31). Figures 3.9 and 3.10 (32) show that this thrust bearing is made integral with the reduction gear casing. The main thrust bearing is usually aft of the gear coupling to the main shaft, but engine room size constraints dictated an integral design.

Of primary concern in the design of this gear is that 50,000 shaft horsepower from a single-input, single-output reduction gear is near state-of-the-art design. Admittedly, the gear tooth loadings are high as shown in Table 3.4, but the design remains feasible as indicated by the General Electric Company.

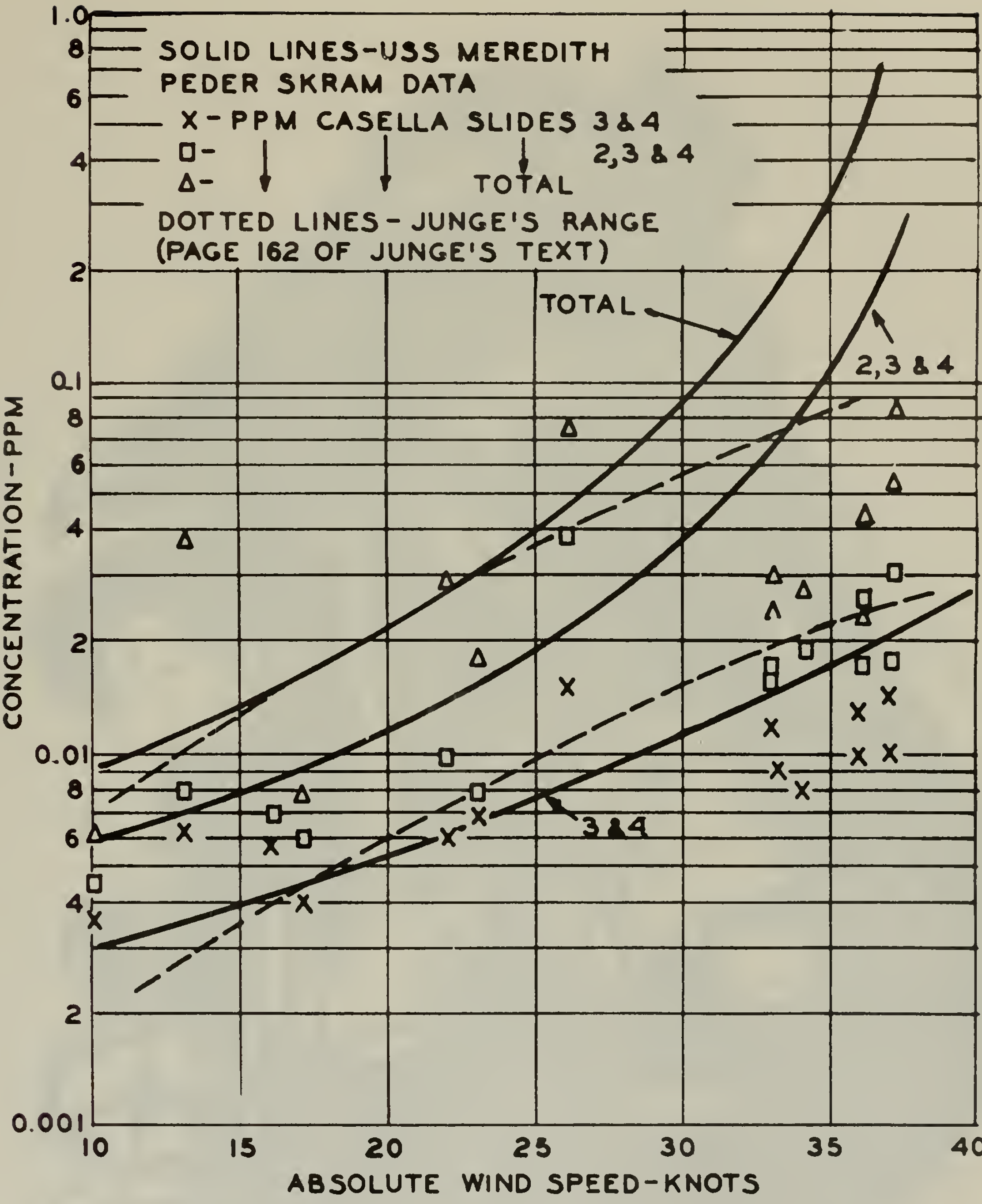


Figure 3.7 Averaged Sea Salt Aerosol Concentrations (30)

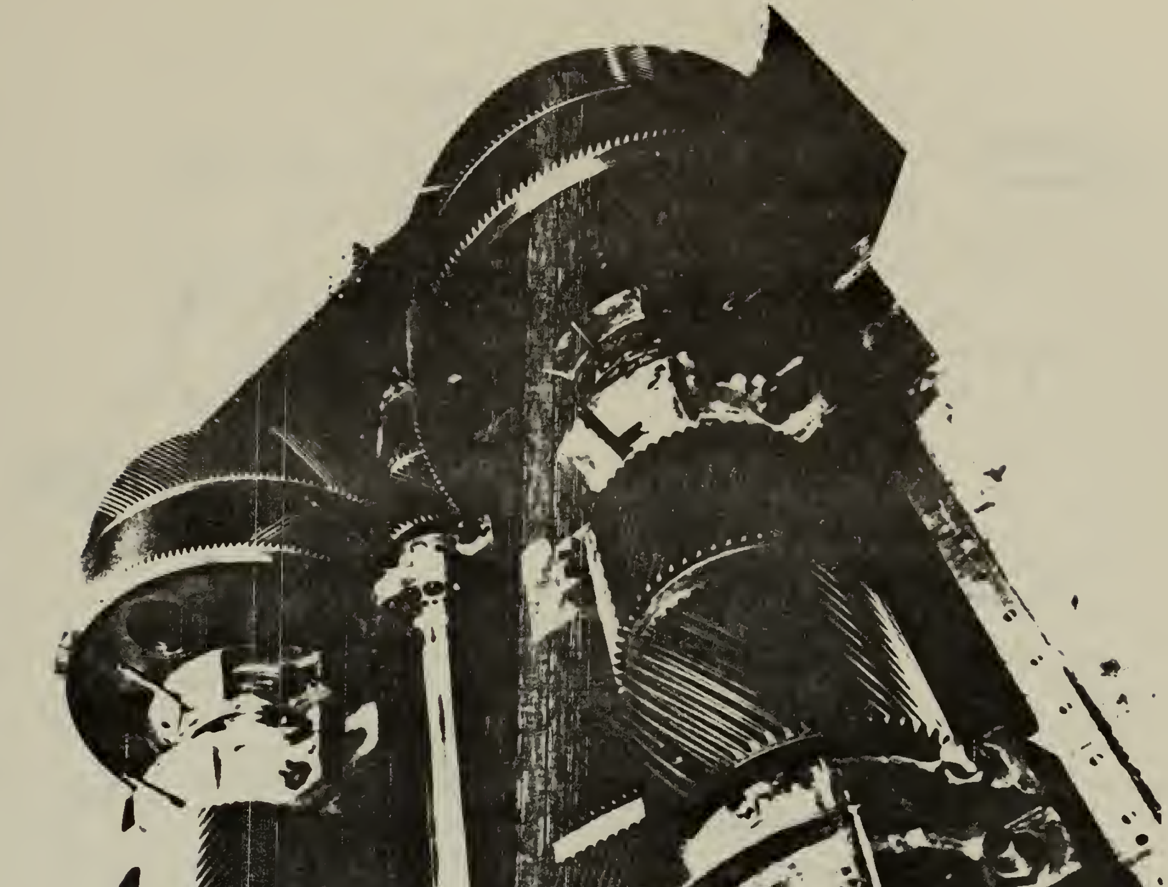
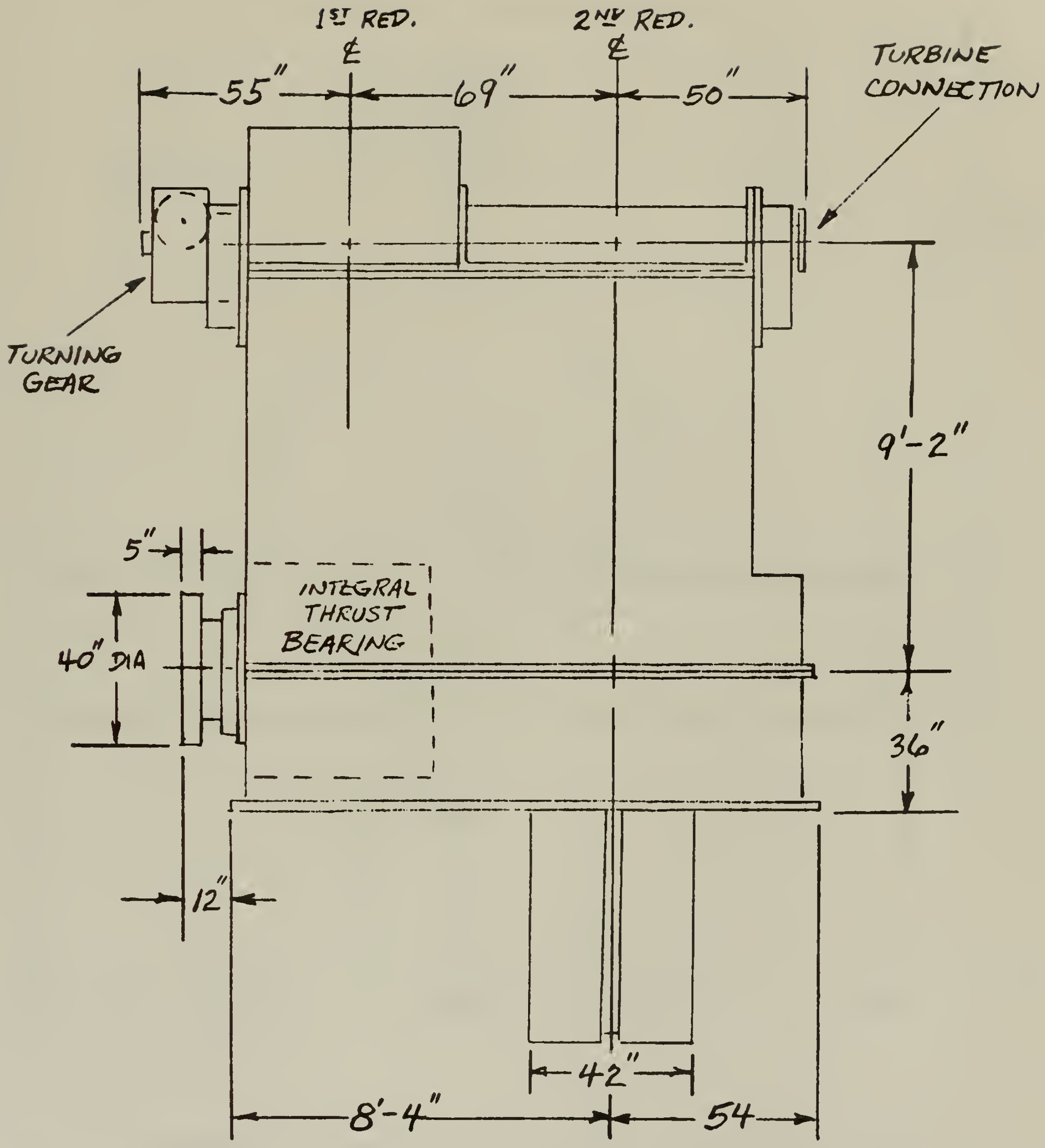


Figure 3.8 Main Reduction Gear (31)



GEAR RATING: 50,000 HP, 4670/140 RPM

EST. WEIGHT: 247,000 LBS.

Figure 3.9 Main Reduction Gear--Elevation (32)

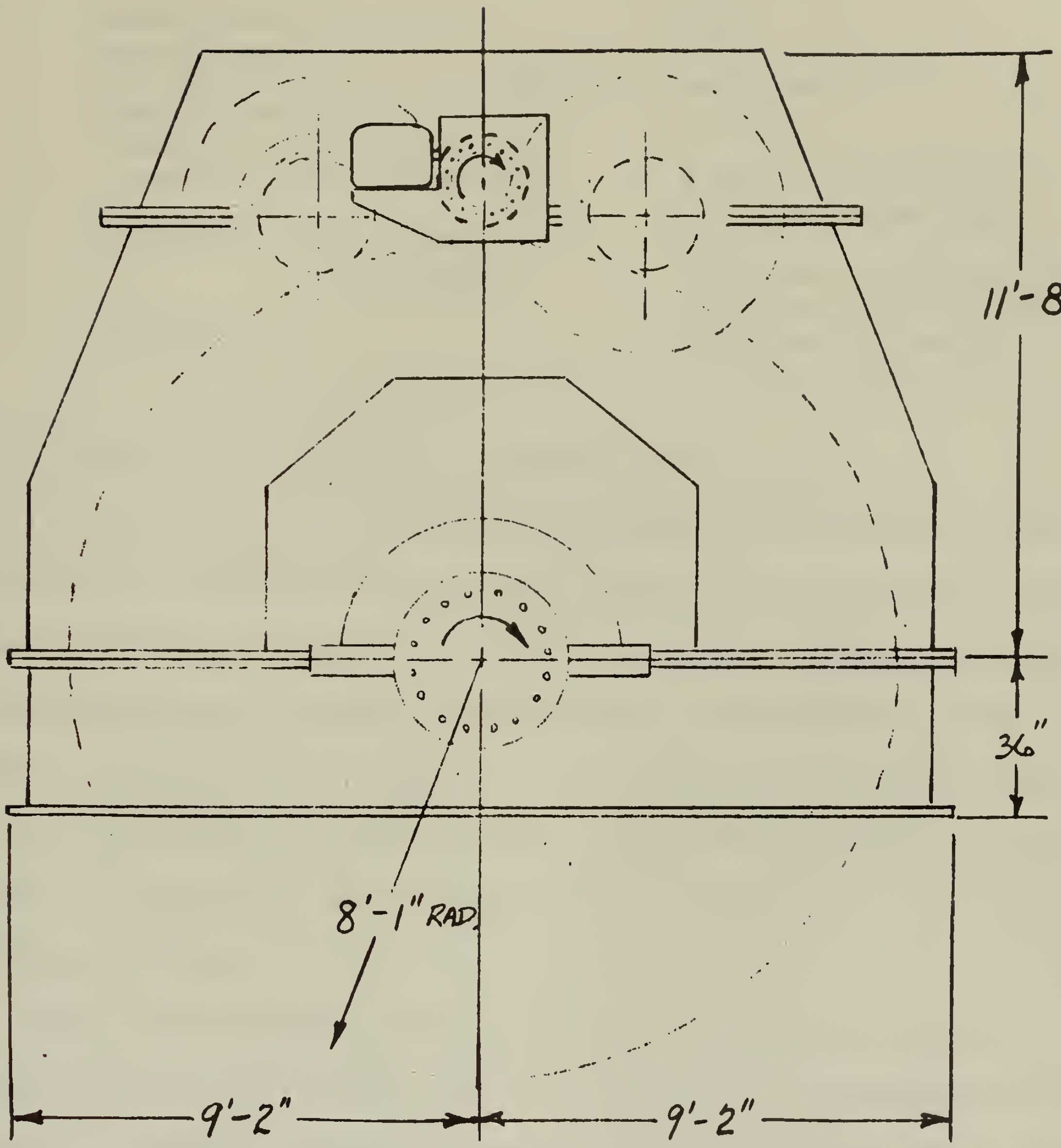


Figure 3.10 Main Reduction Gear--Cross Section (32)

TABLE 3.4 REDUCTION GEAR DESIGN(32)

Design Shaft Horsepower:	50,000
Gear Efficiency:	.985
Reduction Ratio:	33.4:1 (4670/140 at full power)
Lube Oil Requirement:	320 GPM
Loading per inch of gear face per inch of pitch diameter:	High speed rotors 1785
	Low speed rotors 2770
Tooth Root Stress (Dolan-Broghamer)	High speed rotors 20,900 psi
	Low speed rotors 22,800 psi
K Factors:	High speed rotors 150
	Low speed rotors 125

3.2.4-b. Controllable and Reversible Pitch Propeller

A controllable and reversible pitch propeller was chosen to propel the ship as it appeared to be the least complex and most reliable. Choice of other possibilities included reversing gears and/or clutches, a hydraulic transmission system, electric drive or even a reversing gas turbine. Marine controllable pitch propellers, first developed in the mid 1930s have been available for some 35 years. Only recently, however, has there begun to be widespread use of them--both in the numbers in use and the horsepower ratings.

Again, the primary question which must be answered for proof of feasibility is the very large power level at which the propeller will be operating. Largest installations to date are near 30,000 shaft horsepower, and while the U. S. Navy is about to test a 40,000 SHP propeller for the DD-963 class of destroyers, 50,000 SHP does represent a large jump in required power levels. (33) Correspondence with LIPS N.V. (Propellers, Inc.), Sulzer Brothers and Bird-Johnson (KaMeWa) indicate that a controllable pitch propeller at these power levels could be

manufactured and operate successfully. A picture of the KaMeWa design is shown in Figure 3.11 (34). Bird-Johnson Company provided two possible designs at this horsepower range, a four-bladed design and a five-bladed design. (35) The four-bladed design was chosen as it had the smaller diameter--this constraint is dictated by the draft of the ship when lightly loaded. The results of the feasible propeller characteristics appear in Table 3.5 (35).

Due to engine room length constraints, the hydraulic assembly and oil distribution box assembly are located along the shaft instead of directly after the main thrust bearing, which would be most desirable from a maintenance standpoint.

3.3 Steam Generator and Steam System

3.3.1 Waste Heat Recovery Steam Generator

A schematic of the waste heat recovery steam generator (HRSG) or boiler is shown in Figure 3.12 (36). A brief description of this component and its location in the cycle was shown in Section 3.1. Steam generating capability is a function of the exhaust gas flow, the average temperature entering the unit, the steam conditions desired and the thermal effectiveness. See Figure 3.13 (37). The thermal effectiveness is defined as the ratio of total energy recovered to total energy available for steam generation, that is:

$$\text{Effectiveness} = \frac{T_1 - T_2}{T_1 - T_3} \quad (38)$$

Where:

T_1 = Exhaust gas temperature entering the HRSG
T_2 = Exhaust gas temperature leaving the evaporator
T_3 = Saturation temperature at steam drum pressure

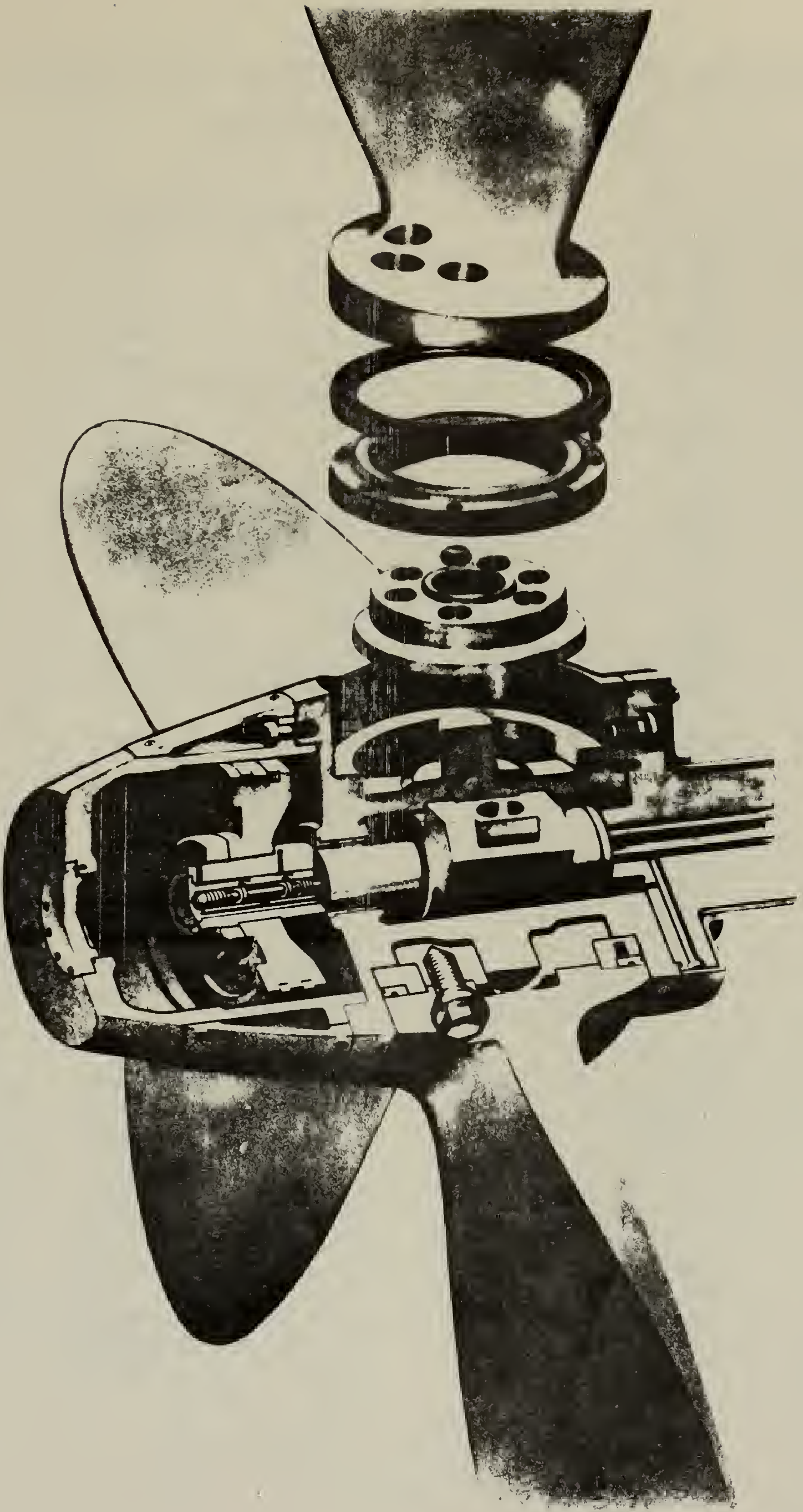


Figure 3.11 $K_aM_eW_a$ Controllable and Reversible Pitch Propeller (34)

TABLE 3.5 (35) 50,000 SHP $K_AM_EW_A$ FOUR-BLADED CONTROLLABLE AND REVERSIBLE PITCH PROPELLER

Number of Blades	4
SHP	50000.
DHP	48500.
RPM	140.
Knots	26.00
Wake	0.100
Prop Diameter (ft)	22.3
Hub	211. S/4
DH/D (%)	31.03
EAR	0.752
P/D, Design	1.031
P/D, Max Forward	1.134
P/D, Max Aft	1.031
Class	DNV
Ice Class	None
Ultimate Strength (psi)	85000.
Blade Thickness Fraction	0.0687
T - Tip (inches)	0.960
T - Hub (inches)	12.97
Reduction 0.7R	0.
Wt - Blade and Flange	19300.
Wt - Propeller	136700.
WR2 - Blade (lb-ft^2)	2720000.
WR2 - Propeller (lb-ft^2)	12710000.
P/D	1.031
ETA Prop	0.624
Thrust (lbs)	415000.
Cavitation Margin	0.174
BP	11.4
Delta	133.5
KQ	0.0300
J	0.759
Sigma 0.85	1.92
Sigma' 0.85	1.11
T/KT	1216482.

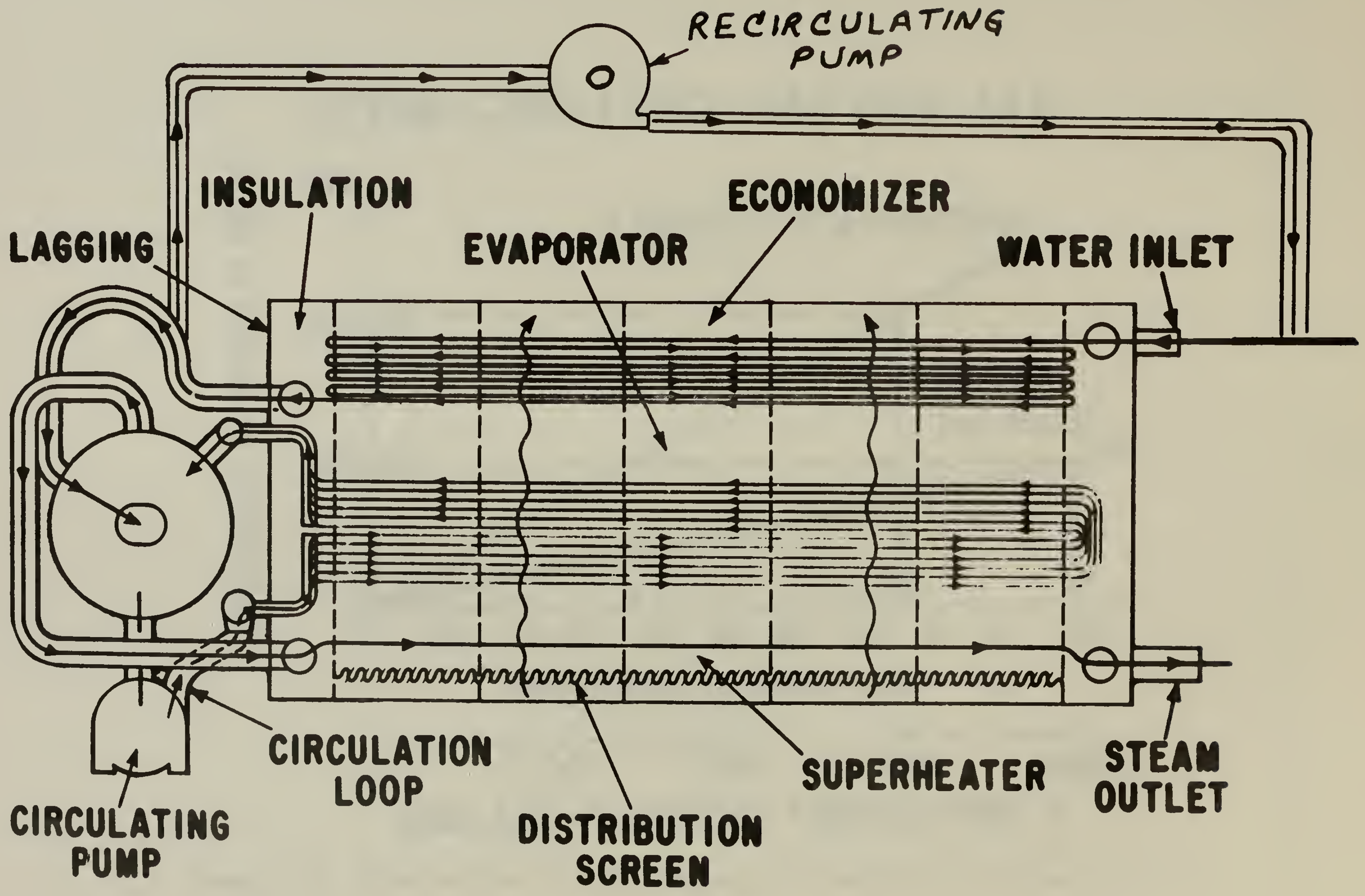


Figure 3.12 Waste Heat Recovery Steam Generator (36)

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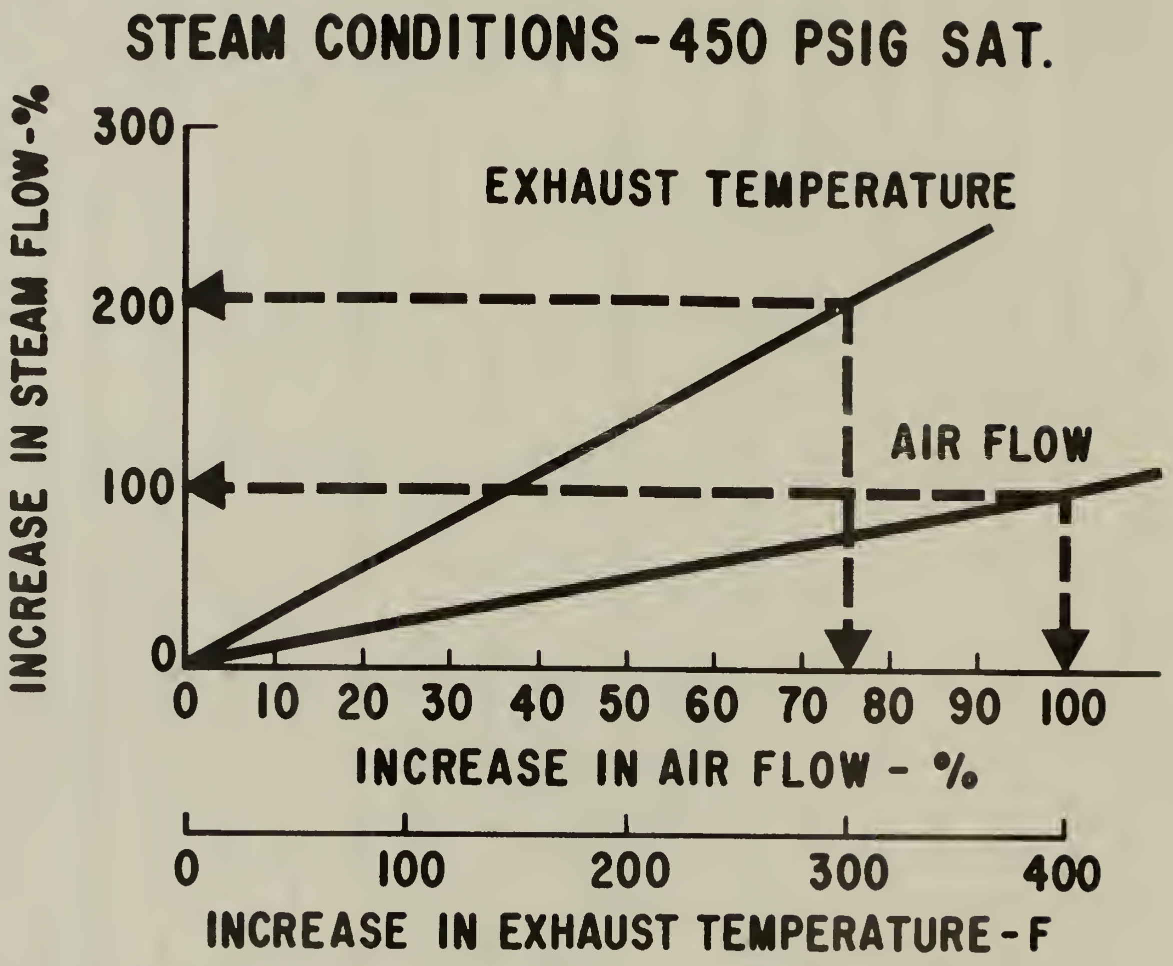


Figure 3.13 Relative Effect of Air Flow and Gas Temperature on Steam Flow (37)

[illegible]

For boiler conditions at full power shown in the supercharged cycle, the effectiveness is 0.875.

The basic thermodynamic principle is best shown in Figure 3.14 (39). Gas turbine exhaust temperature and stack temperatures are plotted as a straight line above the steam generation line, which is broken into three segments--economizer, or preheating, evaporator and superheater. This curve represents the normal 'heat added' portion of the Rankine cycle. Temperature and pressure are raised in the economizer section until the saturation temperature is reached, heat continues to be added (horizontal line) as the saturated liquid becomes saturated vapor, and finally the temperature and pressure rise rapidly into the superheated region. Noteworthy in the diagram is the approach temperature at the beginning of the evaporator section--the so-called "pinch point" which, in this cycle, was 30° F. Initial feasibility designs for this cycle selected a steam pressure level of 600 psig; this was reduced to a drum operating pressure of 475 psig (450 psig line pressure) in order to reduce maintenance costs and lower stack temperature to make a more efficient cycle. The design of a heat recovery steam generator becomes an iterative one in order to optimize the cycle, and the General Electric Company provided output data as shown in Table 3.6 using their Performance Program for Heat Recovery Steam Generators. (40) The desire to lower stack temperature was not realized as the output data shows--a lower steam pressure probably would have brought the stack temperature down and made for a more efficient cycle. The importance of stack temperature in waste heat boilers cannot be overemphasized, as "cold end corrosion" in the economizer section can cause complete boiler failure. This is caused when the sulfur in the fuel

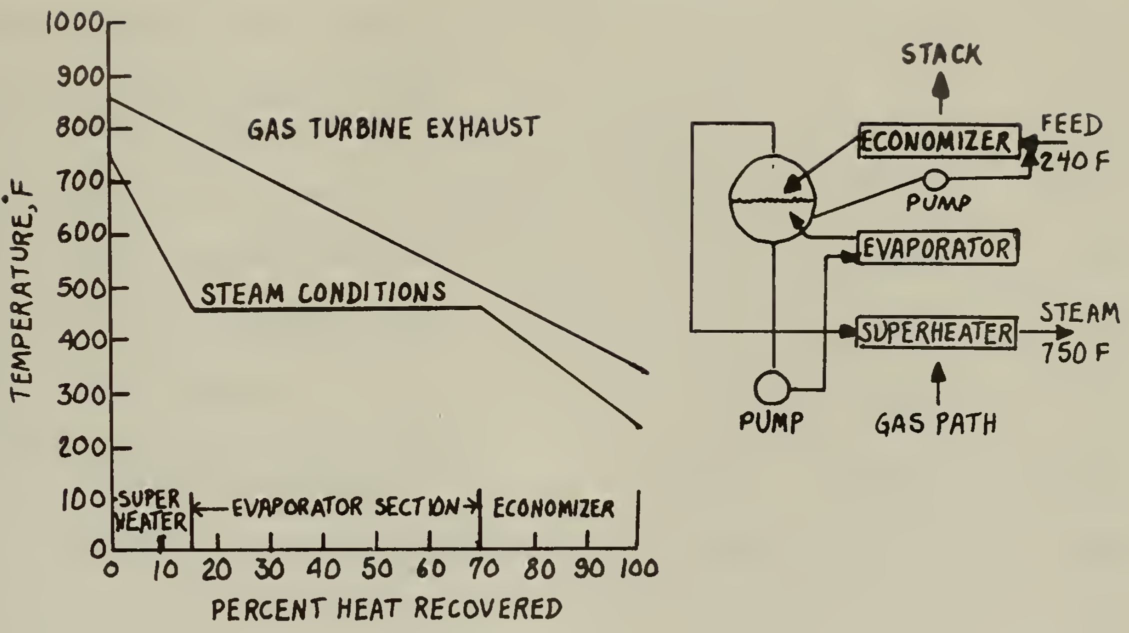


Figure 3.14 Temperature Relationships in Heat Recovery Steam Generator (39)

TABLE 3.6 (40) HEAT RECOVERY STEAM GENERATOR PERFORMANCE

Boiler Pressure (HRSG) 475 psig, 450 psig at throttle

		FULL POWER	17 KNOTS	12 KNOTS
*(1)	Gas Temperature (exhaust) ° F.	853	703.5	624.5
*(2)	Gas Flow (exhaust) lb/hr (3% leakage loss in bypass)	1,252,253	856,000	826,000
(3)	Superheater Steam Flow, lb/hr	102,000	39,650	24,930
(4)	Steam Temperature--Supht ° F.	750	680	620
(5)	HRSG Back Pressure, "H_2O	9"	4.4"	4.1"
(6)	Stack Temperature, ° F.	355	375	400
(7)	Recirculation Flow, lb/hr (through economizer)	96,000	43,000	27,000
(8)	Gas Temp., Leaving Superheater,°F.	800	680	615
(9)	Gas Temp. Leaving Evaporator, ° F.	510	480	475
*(10)	Saturated Steam Flow, lb/hr	15,000	15,000	10,500

*Given inputs

condenses with water at the dewpoint temperature and forms sulphuric acid, which attacks the tubes. A diagram of this relationship is shown in Figure 3.15 (41). Since Navy distillate has a maximum sulfur content of 1.3%, the stack temperature could have been reduced to 270° F. In any event, this was not done, and the stack temperature at full power remains a conservative 355° F.

A formal description of the waste heat boiler would call it a vertical gas flow forced circulation unit with three separate heat transfer sections. (36) Heat is transferred by forced convection with the various sections arranged in counterflow. The evaporator and economizer sections are fitted with circulation pumps. Forced circulation in the evaporator section (Figure 3.12) allows the vertical gas installation and makes a more compact steam generator. It also maintains constant flow to the evaporator over the full range of steam output, eliminating the possibility of stagnation in the evaporator section. The economizer is fitted with a recirculation pump to raise the feed temperature and inlet from 105° F. to 240° F. This is done to prevent corrosion in the economizer and ensure longer life.

The heat transfer surface is mounted inside a frame made of 3/4 inch plate. All tube return bends and the various headers associated with each heat transfer section are located in casing boxes outside the gas flow container. The casing boxes are welded to the unit frame resulting in a gas-tight container. The economizer and evaporator use extended finned tubes. The fins are continuously wound and welded on the tube in a single operation. Evaporator and economizer tubes are made of carbon steel, one inch in diameter, wall thickness of .095 in. and a fin outside diameter of

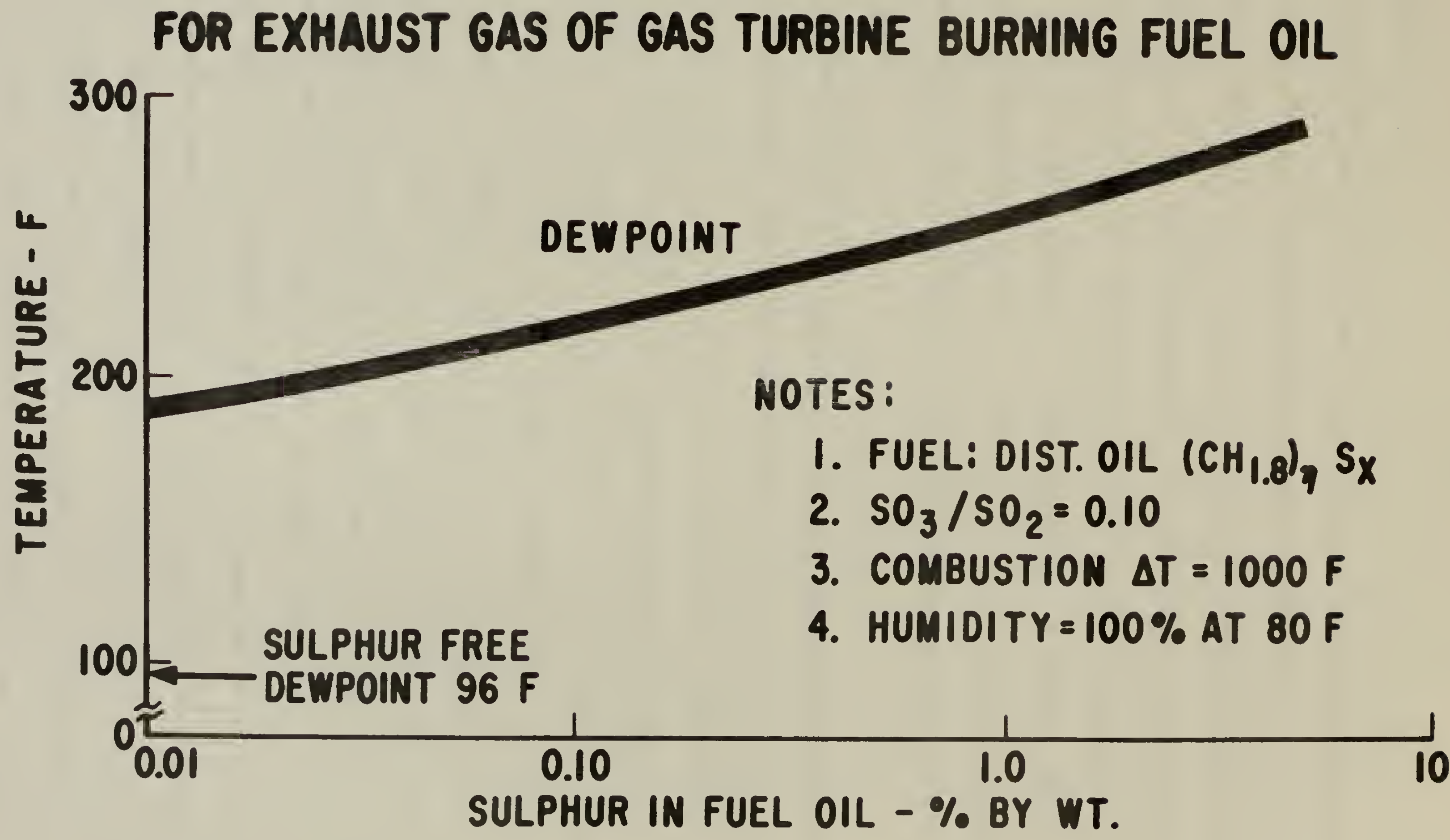


Figure 3.15 Acid Dewpoint and Condensation Temperatures (41)

2.0 inches.

The steam drum is of conventional design. Internal distributions systems are provided for the feedwater, water sampling and chemical feed. Centrifugal separators are located in the drum to reduce the total solids.

The size of the waste heat boiler for this cycle has external dimensions of 11 ft. by 42 ft. by 12 ft. in height. The gas passage is 9 ft. by 35.5 ft., with duct sizes of 10 ft. by 12 ft. A by-pass duct to allow for underway maintenance is 10 ft. by 10 ft. The heat recovery steam generator weighs 170,000 pounds dry and 210,000 pounds when filled with water. (40)

3.3.2 Steam Driven Supercharger, Turbogenerators and Feed Pumps

The supercharger compressor is directly driven by a steam turbine with throttle pressure of 450 psig. The turbine is a twelve-stage, single-flow, impulse turbine with a higher reaction last stage. (42) The steam rate (lbs steam/BHP-hr) is 6.95 and the turbine expands directly into the main condenser at 2" Hg absolute pressure. The turbine will be able to operate at varying steam pressures as the steam cycle is a pressure controlled cycle, that is, line pressure is held constant at 450 psig for the turbogenerators and feed pumps. Steam supply and demand is controlled by a pressure regulating valve, which may admit steam to the supercharger turbine or dump it to the main condenser via breakdown orifices and nozzles.

Each engine room will contain two ship's service turbogenerators rated at 1500 KW, 450 volt, 3-phase, A-C static excited, static voltage controlled similar to those presently installed in the AOE-1. The multistage turbine will receive steam at full superheater outlet pressure and temperature and exhaust to the main condenser. The turbine is provided with sequential

valve gear and an integral speed governor. A single reduction gear couples the turbine to the generator.

The main feed pumps (two) are horizontal two-stage centrifugal direct connected and driven by superheated steam at 450 psig. Each of the pumps is designed to handle all of the feed water requirements of the waste heat boiler at full power, delivering feed water to the economizer at 700 psig. The steam turbine is a single-stage unit designed to supply steam for the maximum rating of the pump, and steam is exhausted directly into the main condenser. The main feed pumps are provided with a pneumatically operated constant pressure governor. An automatic recirculating valve will close when the pump capacity exceeds approximately half the normal rating.

3.3.3 Vital Auxiliaries

3.3.3-a. Main Condenser and Condensate and Feed System

The main condenser characteristics and size may be found in Table 3.7. As summarized from Appendix A, the main condenser receives steam from the supercharger turbine, the turbogenerator sets, and the main feed pumps. It must also be capable of receiving 'dump' steam depending upon the supply and demand of the plant. Low pressure drains are admitted directly to the hotwell, as shown in Figure 3.1. The condenser is single pass, designed for scoop circulation with water velocity in the tubes of 9 ft. per second at normal power. Tube and tube sheet material is 90-10 copper-nickel. The shell is of fabricated steel, and tube inlet ends are fitted with plastic inserts. The main condenser must be able to deaerate the water completely by using deaerating trays and baffles under the tube bundle, similar to the DeLaval design used aboard submarines. Additionally,

TABLE 3.7 CONDENSER CHARACTERISTICS (FULL POWER)

(See Appendix A)

Tube Size:	5/8" 18 Gauge
Material:	90-10 Copper-nickel
Number of Passes:	1 - Scoop injection
Vacuum:	2" Hg (1.0 psia)
Effective Tube Length:	10 feet 3 inches
Steam Rate (max):	117,000 lbs/hr
Injection Temperature:	67° F.
Inlet Water Velocity:	9 feet/second
Circulating Water Rate:	14,500 G.P.M.
Condensing Surface Area:	4000 ft^2
Heat Transfer Coefficient:	615 Btu/hr-ft^2 ° F.
Number of Tubes:	2,370
Water Box Depth:	2 feet 7 inches
Condenser Length:	15 feet 6 inches
Condenser Width:	6 feet 5 inches
Condenser Height:	6 feet 5 inches
Hotwell Capacity:	31.4 ft^3/min

the condenser must be capable of maintaining vacuum when large quantities of steam are dumped. This is accomplished via multiple breakdown orifices and nozzles within the shell. This practice is used widely in industry without adversely affecting the steam cycle.

Vacuum is maintained with two mechanical and automatic vacuum pumps instead of the usual steam air ejector system. They are complete with motor, seal tank, air ejector and salt water-cooled heat exchanger. The cycle uses two motor driven main condensate pumps which are mounted on and supported by the main condenser. Each pump will have the capacity to pump all condensate at maximum power with the other pump as a standby. Discharge pressure to the suction side of the main feed pump is 60 psig.

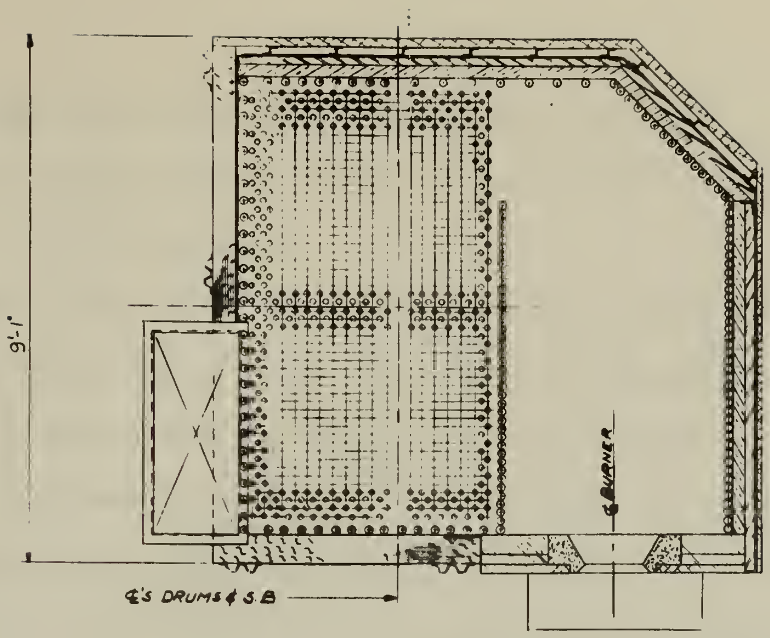
Details of the specifications for the above components and other auxiliaries associated with the feed and condensate system may be found in Appendix D., List of Equipment and Specifications.

3.3.3-b. Auxiliary Boiler

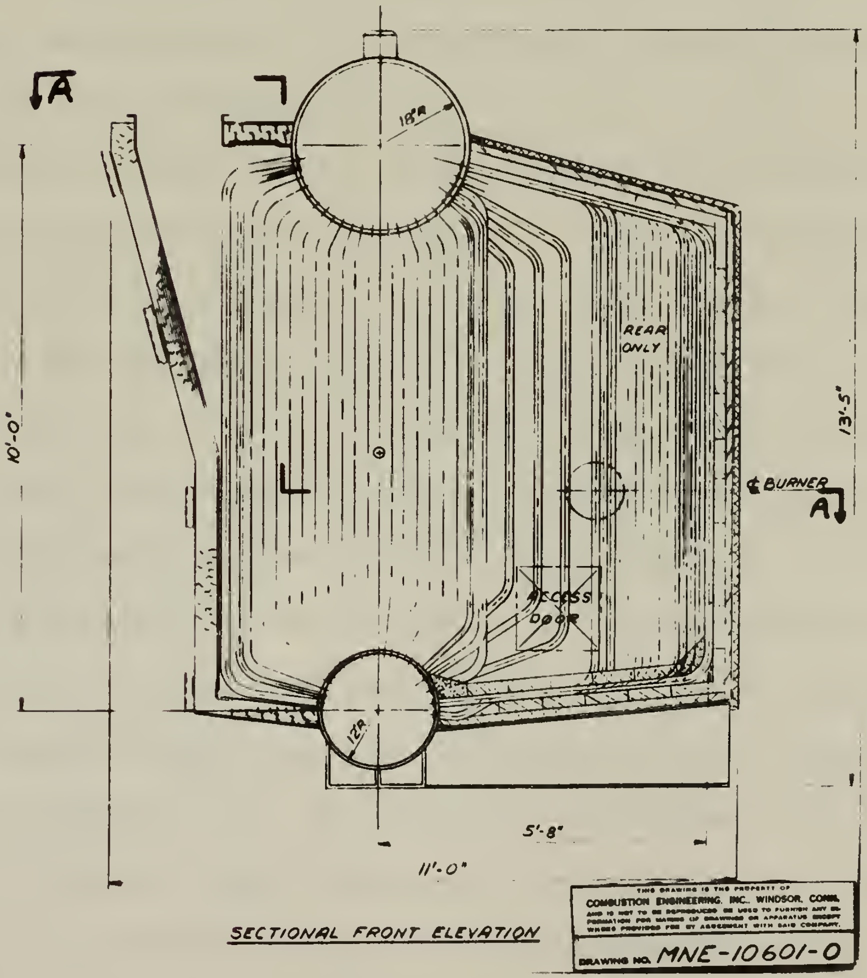
The auxiliary boiler for use when not running the main steam cycle is shown in Figure 3.16 (43). It is of the two-drum welded waterwall type with a furnace section, a welded screen tube bank, and a main bank. The boiler is rated at 135 psig and 30,000 lbs/hr. The boiler is a Combustion Engineering V2M design, with characteristics as listed in Table 3.8 (43).

TABLE 3.8 (43) Auxiliary Boiler Characteristics

Steam Drum Pressure:	135 psig
Total Evaporation Rate (Saturated)	30,000 lbs/hr
Total Steam Temperature:	358° F.
Efficiency:	80%
Oil Firing Rate:	2075 lbs/hr
Air Flow and Temperature	7,790 CFM @ 100° F.
Exhaust Gas Flow:	35,300 lbs/hr
Total Draft Loss:	7.7 in. water
Excess Air:	15%
Furnace Release Rate:	155,300 Btu/hr-ft^3



SECTION "A-A"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SECTIONAL FRONT ELEVATION

Figure 3.16 Auxiliary Boiler Design (43)

3.4 Power Plant Control and General Plant Operation

3.4.1 General Method of Control at Full Power, No Load and Full Astern (44)(45)

The operation of the supercharged cycle using a heavy-duty gas turbine may be looked at as the control of two separate but yet integrated systems--the gas turbine section and the steam cycle. Figure 3.17 (46) shows the schematic for control of the plant. The dotted line horizontally dividing the schematic indicates the approach of looking at each section separately. The General Electric Company uses a Speedtronic control system to operate its gas turbines--the control approach that follows describes this method since their gas turbine was selected for the analysis. Controls have been modified on the steam cycle to operate the supercharger compressor, turbogenerators and main feed pumps.

Looking at Figure 3.17, it may be seen that the gas turbine is controlled by manipulating two parameters: 1) fuel flow to the combustors, and 2) the angle of setting of the variable angle nozzles. Fuel flow output is proportional to the product of compressor speed and an electrical voltage, termed VCE. The VCE voltage is produced by four parallel electronic control loops operating through a minimum gate valve. The first loop is the exhaust temperature control loop which is able to limit the VCE when the exhaust temperature reaches its maximum allowable value for the existing conditions. The second loop is the load or low pressure turbine speed loop which reacts similarly when the load turbine reaches rated speed setpoint, i.e., it becomes the controlling variable. The third loop is the start-up loop, which regulates the turbine during start up when the other input parameters (exhaust temperature and LP turbine speed) are

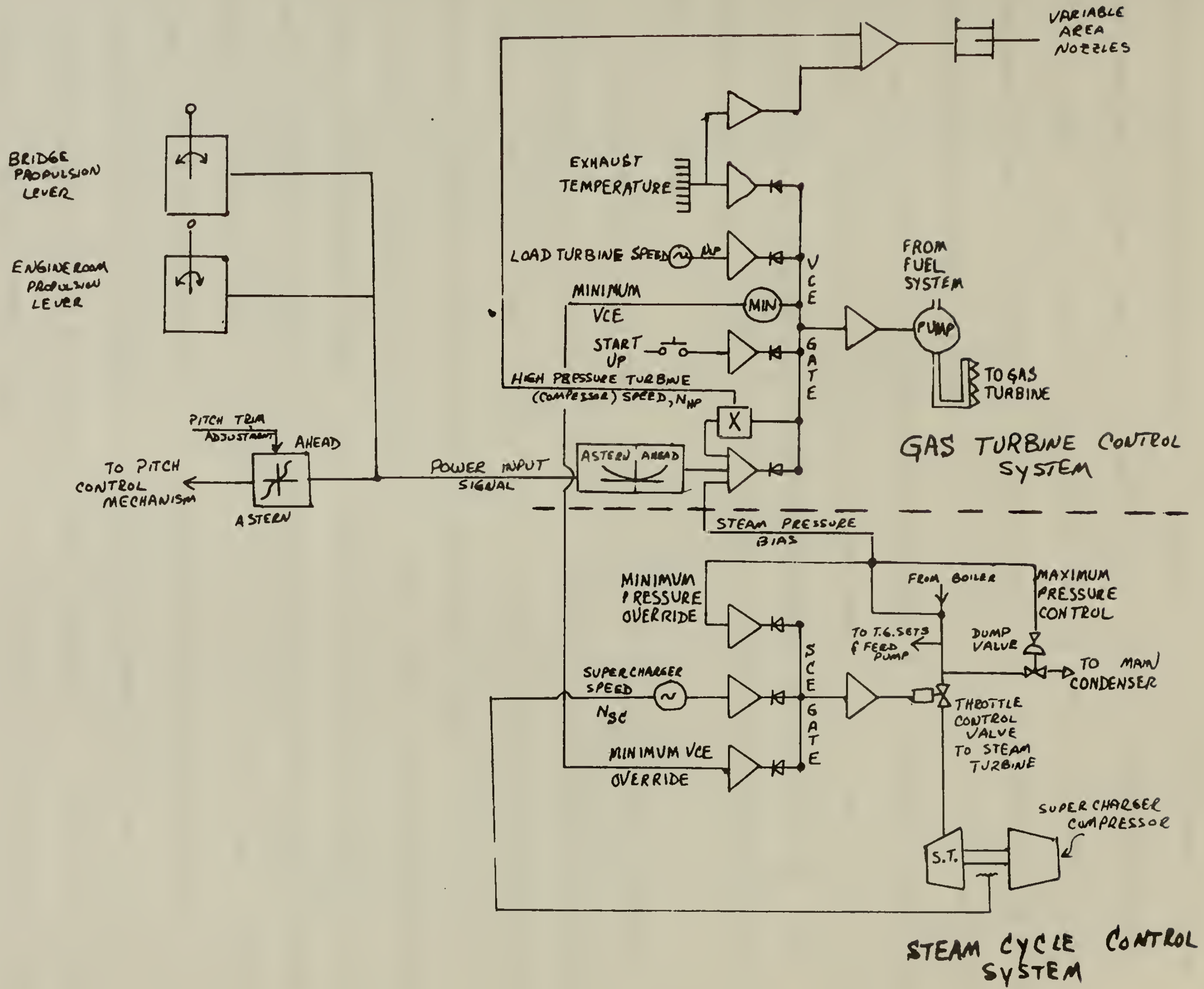

Figure 3.17 Gas Turbine--Steam Cycle Control System (46)

not limiting factors.

The fourth loop, compressor speed, is necessary because if the fuel flow to the two shaft gas turbines is constant, the output power of the low pressure turbine remains the same and independent of low pressure turbine speed. Fuel flow is proportional to the product of the VCE and compressor speed. In order to control power, then, the fourth loop is added which has the power command (which may include pitch) signal as a setpoint and the product of compressor speed and VCE as feedback. See Figure 3.18 (47). The action of this loop is the same as the others; if its output is lowest, it becomes the dominant parameter controlling VCE. The minimum power bias provides the VCE required to keep compressor speed at 80% as a minimum.

Looking again at the temperature control loop again at the top of the schematic, it is recalled that the variable angle nozzles are controlling the compressor speed. Since exhaust temperature is sensitive to the mass rate of flow of air (related to compressor speed), it is possible to hold exhaust temperature at the desired level (just below the maximum) by controlling compressor speed. This enables the two-shaft machine to give good part-load fuel rates. Operating characteristics using this control system are shown in Figure 3.19 (48). From no load to 45% load, compressor speed is held at a minimum of 80% using the variable angle nozzles. At the same time, firing temperature and exhaust temperature are rising as fuel flow increases and the fuel/air ratio increases. Past the 45% load rating, the exhaust temperature exceeds the nozzle control setpoint, the nozzle loop increases the high pressure turbine energy and compressor speed and thus the exhaust temperature lowers back to the setpoint after this

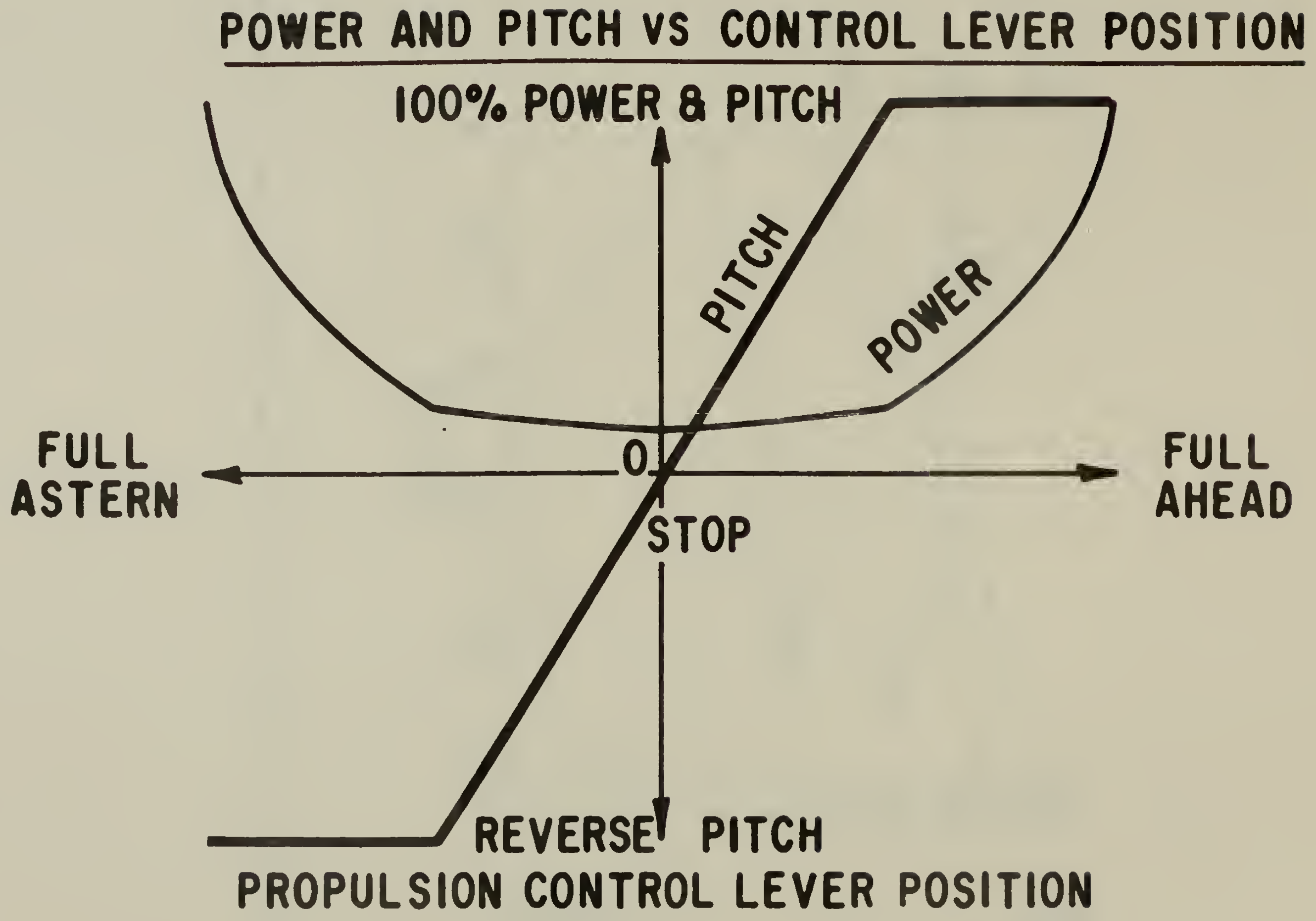

Figure 3.18 Propulsion Control Schedule (47)

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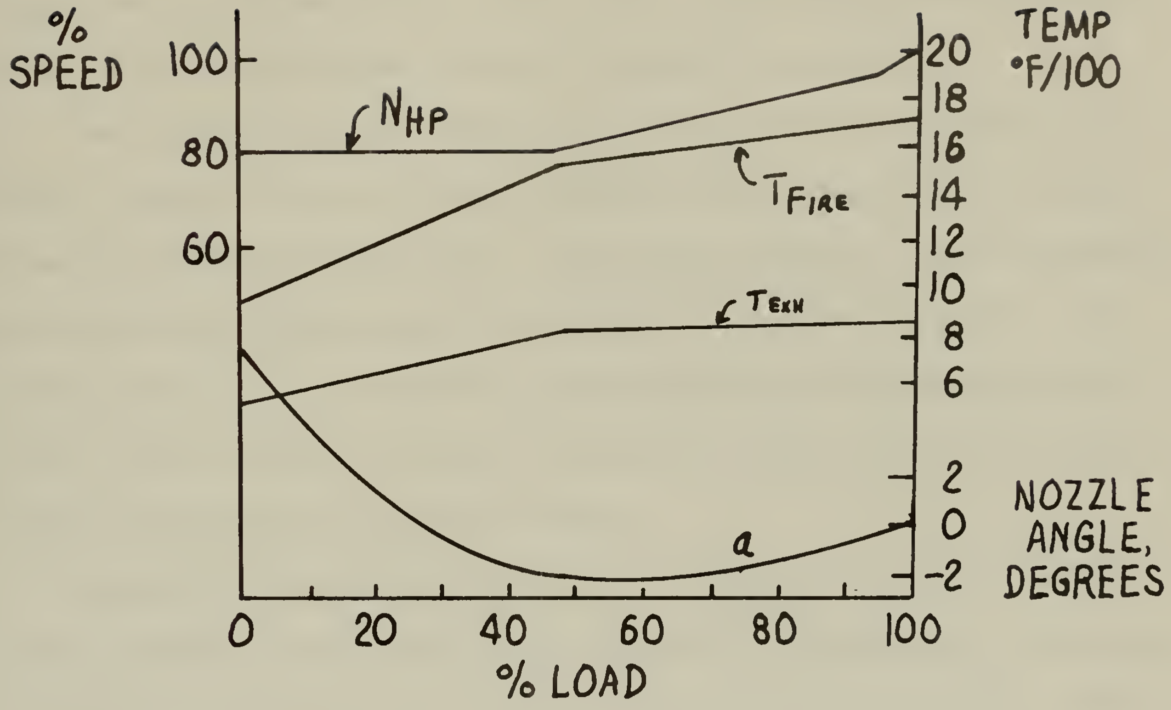


Figure 3.19 Operating Characteristics of Two Shaft Gas Turbine (48)

transient has passed through the control loops. Up to full load, the steady state exhaust temperature remains constant and compressor speed increases with higher fuel flow. When the compressor reaches 100% speed, the control limit is reached and fuel flow tends to increase along with exhaust temperature. The increased exhaust temperature decreases the exhaust temperature fuel control loop output voltage below the rest of the other loops--it therefore becomes the controlling variable.

The lower half of the control schematic shows the steam cycle controls. Not shown are the normal standard control systems for the HRSG drum level and other auxiliaries. The basic component to be controlled is the supercharger speed since it is related to gas turbine power. The basic philosophy in the control of the steam cycle is pressure control, that is, line pressure is held at 450 psig to supply steam at 450 psig to the T-G sets and the main feed pumps.

Figure 3.17 also shows an SCE gate, analagous to the gas turbine control system. It has three input loops and one output loop, which is the position command for the steam turbine control (steam throttle and condenser dump valves). The first input limit, called minimum pressure override, regulates steam flow from the waste heat boiler to ensure that 450 psig line pressure is held. The second loop is supercharger speed control, which becomes the controlling variable if supercharger speed reaches 100%. The third loop is the minimum VCE from the gas turbine control circuit which allows the gas turbine to overide steam flow in order to provide minimum input to the propeller by changing steam to the supercharger compressor and therefore airflow to the gas turbine. The coupling between the two control systems is the power/steam line pressure

connection. If steam line pressure is made an input to the gas turbine fuel flow signal at the power control loop, then total input power to the propeller is regulated.

To look at the actual response of the gas turbine, we shall take the case of the ship underway at full power, then a speed reduction to 45%, or 21 knots, approximately 100 RPM. The power lever on the bridge is reduced introducing a signal to the VCE gate. Note that propeller pitch remains fixed at a maximum until about 45% power, thus a change in power input does not necessarily mean a reduction in propeller pitch. Fuel flow is now cut back by the load turbine governing system, i.e., variable angle nozzles. The variable angle nozzles begin to close, slowing down the compressor speed, yet holding exhaust temperature constant. As compressor speed slows down, gas flow to the heat recovery steam generator goes down, and the boiler line pressure tends to fall. The steam line pressure bias tends to close the dump valve to the main condenser; however, the VCE input to the SCE gate tends to also close down the steam valve to the supercharger to reduce supercharger speed, pressure ratio, and air flow going to the main gas turbine. This signal, therefore, overides the dump valve control, which then readjusts to keep line pressure constant during this transient condition. Further reduction in power changes the propeller pitch, and compressor speed reaches its minimum value at 80% rated speed. The variable angle nozzles now begin to open, fuel flow is further reduced, lowering exhaust temperature. The steam pressure falls off with reduced exhaust temperature, the supercharger control valve closes to reduce supercharger speed, and the dump valve settles out at an equilibrium setting to maintain line pressure. At very low power levels, the supercharger control

valve would be closed off completely (except for cooling steam to the turbine) and the dump valve would be nearly fully closed to keep line pressure constant. It is anticipated that both turbogenerators and feed pumps can function with steam pressures as low as 300 psig, and while this is not desirable, the plant would still be able to perform.

At the condition of full power astern, the system sees the same conditions as at full power ahead except for the fact that the propeller thrust is in the full astern position. Figures 3.20 and 3.21 (49)(50) show what the bridge control and engine room control panels might look like. While control would normally be on the bridge, the engine room would have a positive overide capability if emergency conditions warranted such an action. The typical sizes and weights of the control panels are shown in Figure 3.22 (24) and a typical annunciator/indicator section of the engine room control panel is shown in Figure 3.23 (24).

3.4.2 17-Knot Endurance and 12-Knot Replenishment Plant Ratings

Table 3.1 indicated the critical parameters in the supercharged cycle for all conditions investigated, i.e., full power, 17-knot endurance speed and 12-knot replenishment speed. Figures 3.24 and 3.25 show these off-design power plant ratings. The method of analysis for these conditions may be found in Appendix A., Thermodynamic Considerations of the Cycle.

Figures 3.26 and 3.27 (51) show AOE-1 class shaft horsepower, ship speed and shaft RPM relationships.

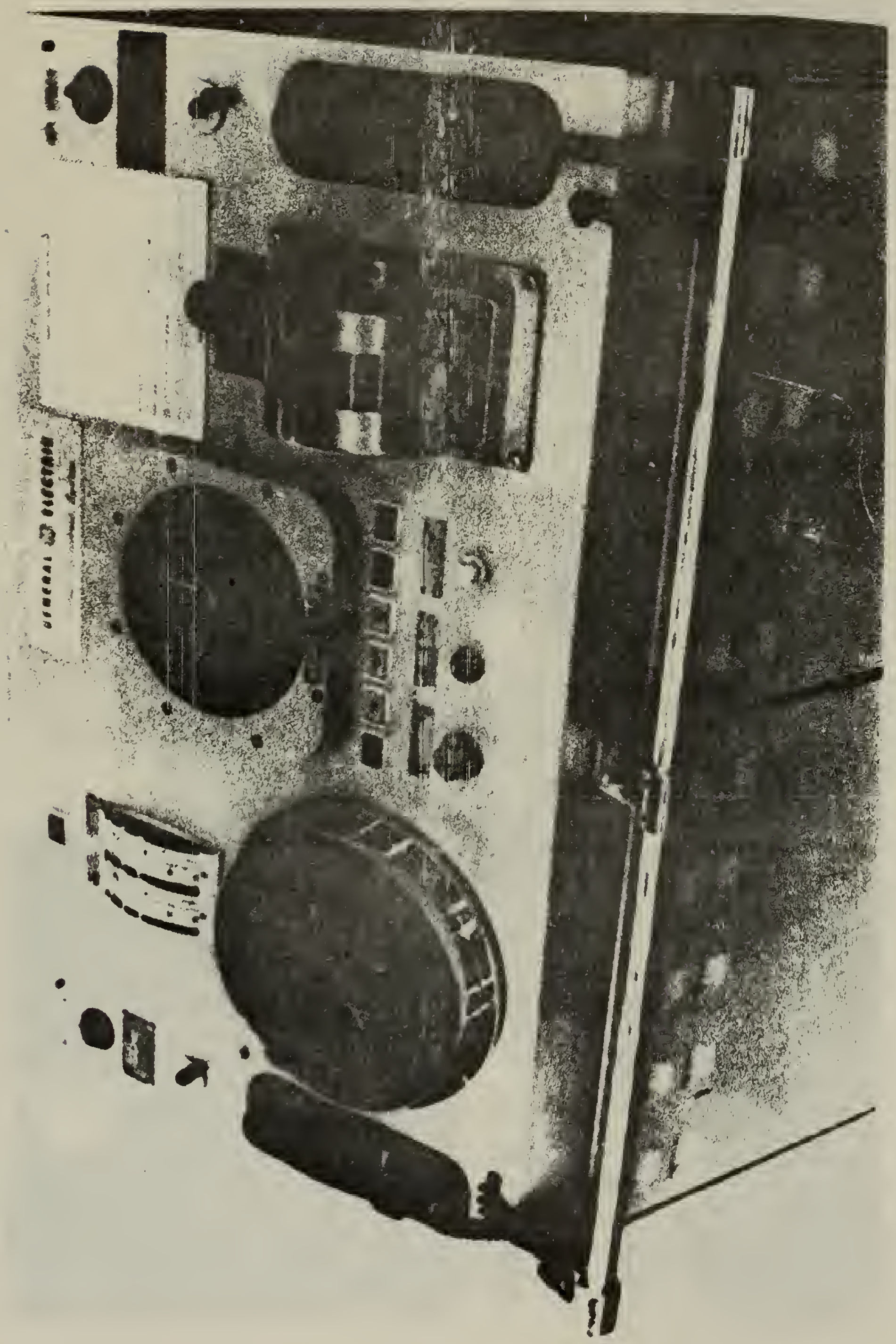


Figure 3.20 Bridge Console (49)

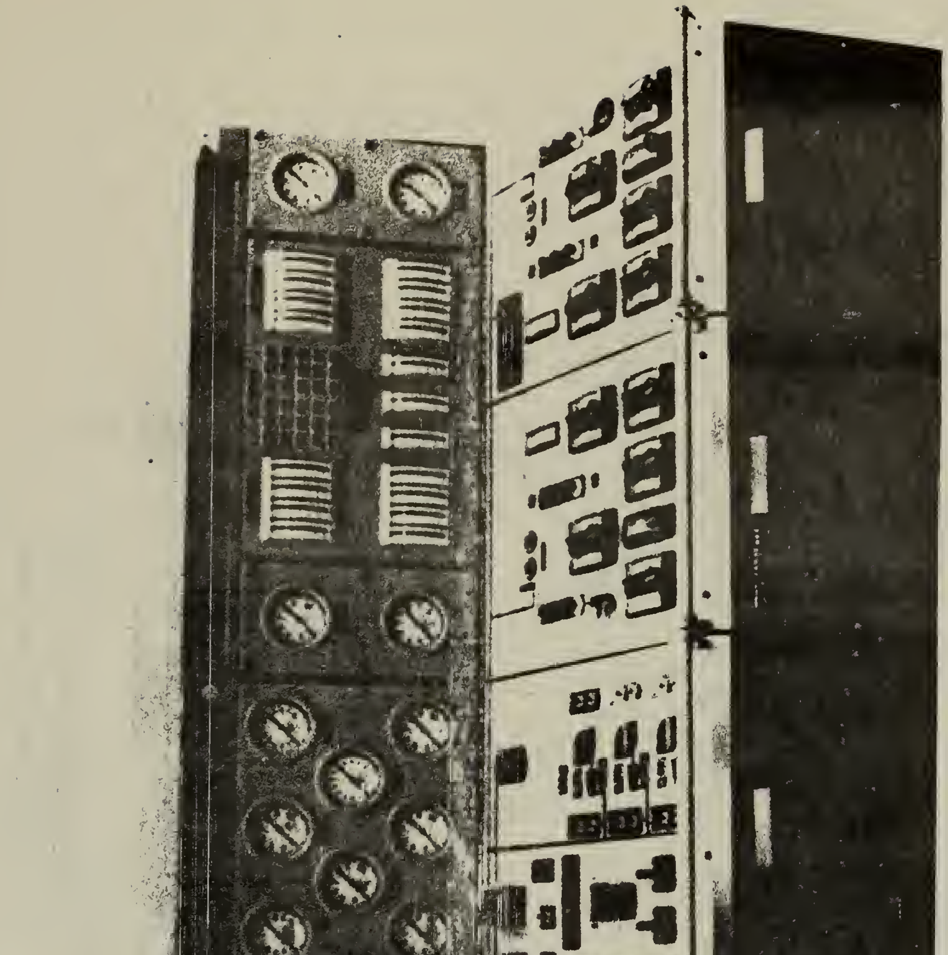
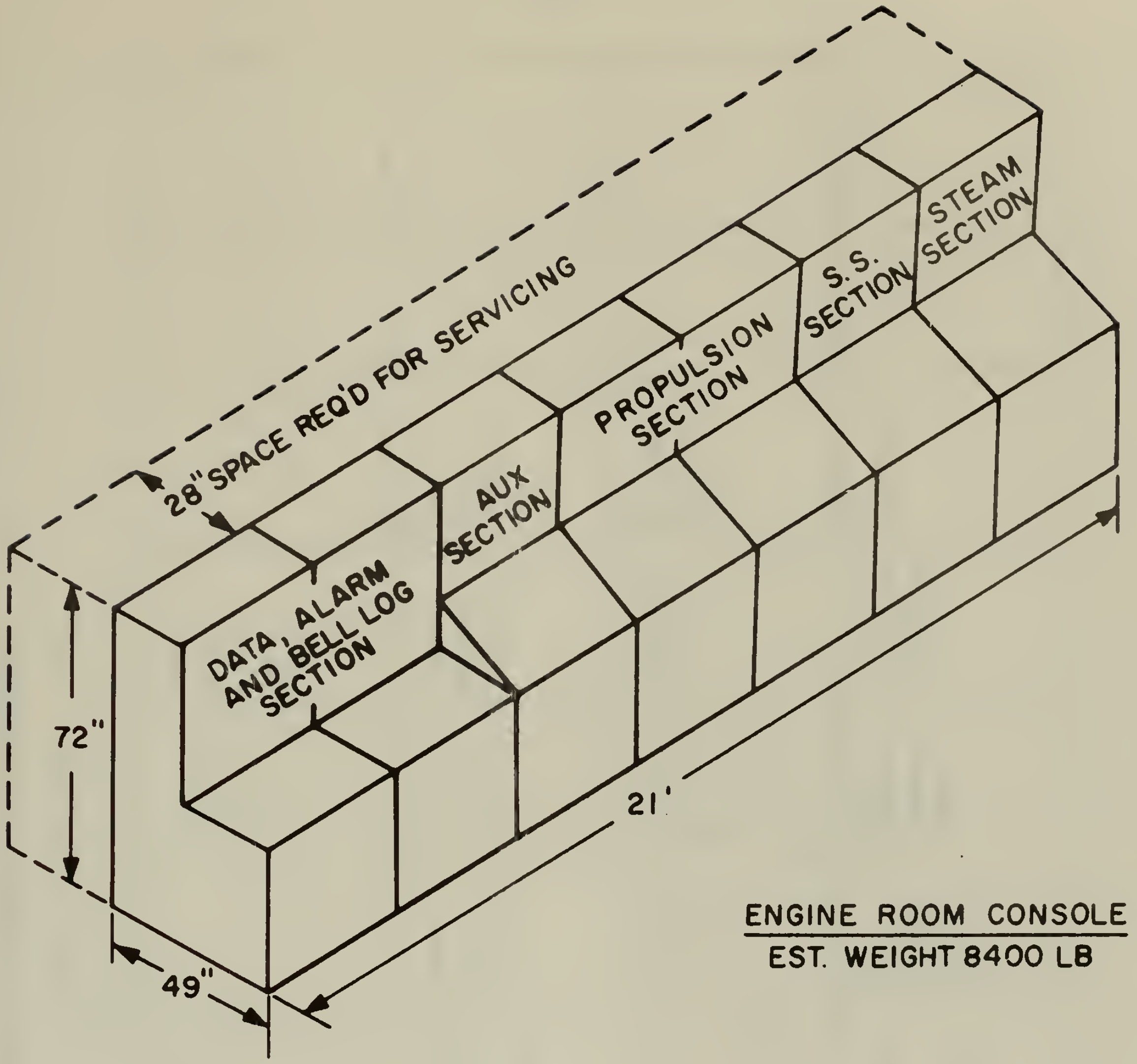
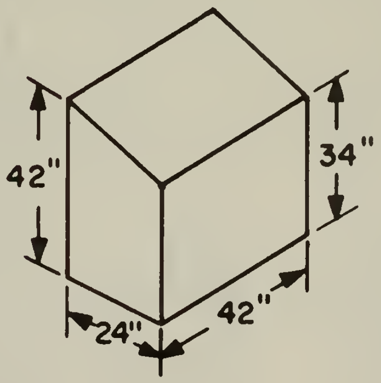


Figure 3.21 Engine Room Console (50)



ENGINE ROOM CONSOLE

EST. WEIGHT 8400 LB



BRIDGE CONSOLE

EST. WEIGHT 500LB.

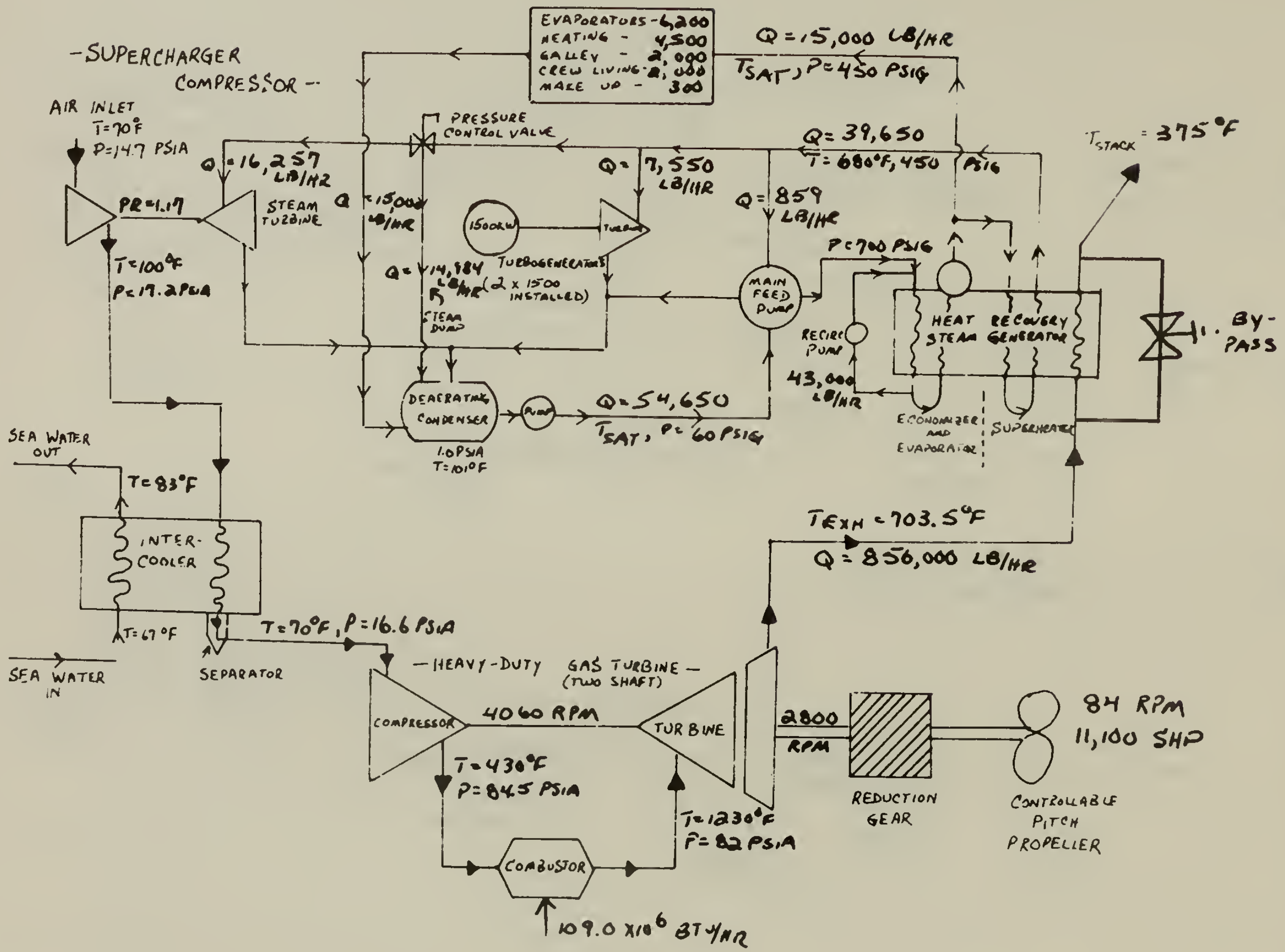
Figure 3.22 Bridge and Engine Room Consoles, Sizes and Weights

Figure 3.23 Standard - Annunciator Arrangement

Two Shaft - Marine Drive (24)

1 Motor Control Center Under Volt	2 Turbine Air Inlet Trouble	3 Cooling Water Pump Not Running	4 Lube Oil Level High or Low	5	6 Starting Clutch Fail to Engage	7 Battery or dc Motor Under Volt	8	9	10
11 Aux. Motor Overload	12 Power Supply Trouble	13	14 Emergency Lube Oil Pump Running	15 Speed Circuit Abnormal	16 Position Servo Trouble	17 Control System Grounded	18 * Diesel Fuel Pressure Low	19	20
21 * Starting Diesel Trouble	22 Control Card Removed	23 Cooling Water Temp. High	24 Lube Oil Header Temp. High or Low	25 Incomplete Sequence	26 Exhaust Temp. System Trouble	27 Vibration Trouble	28 Fuel Oil Pressure Low	29	30
31 Fire	32 Station Emergency Tripped	33 Aux. Lube Oil Pump Running or Low Lube Oil Press.	34 Hydraulic Trip Circuit Pressure Low	35 Fail to Fire or Loss of Flame	36 Exhaust Temp. High	37 Excessive Vibration	38 Overspeed	39	40

*If used



SUPERCHARGED GAS TURBINE
AND STEAM PLANT

- 17 KNOTS -

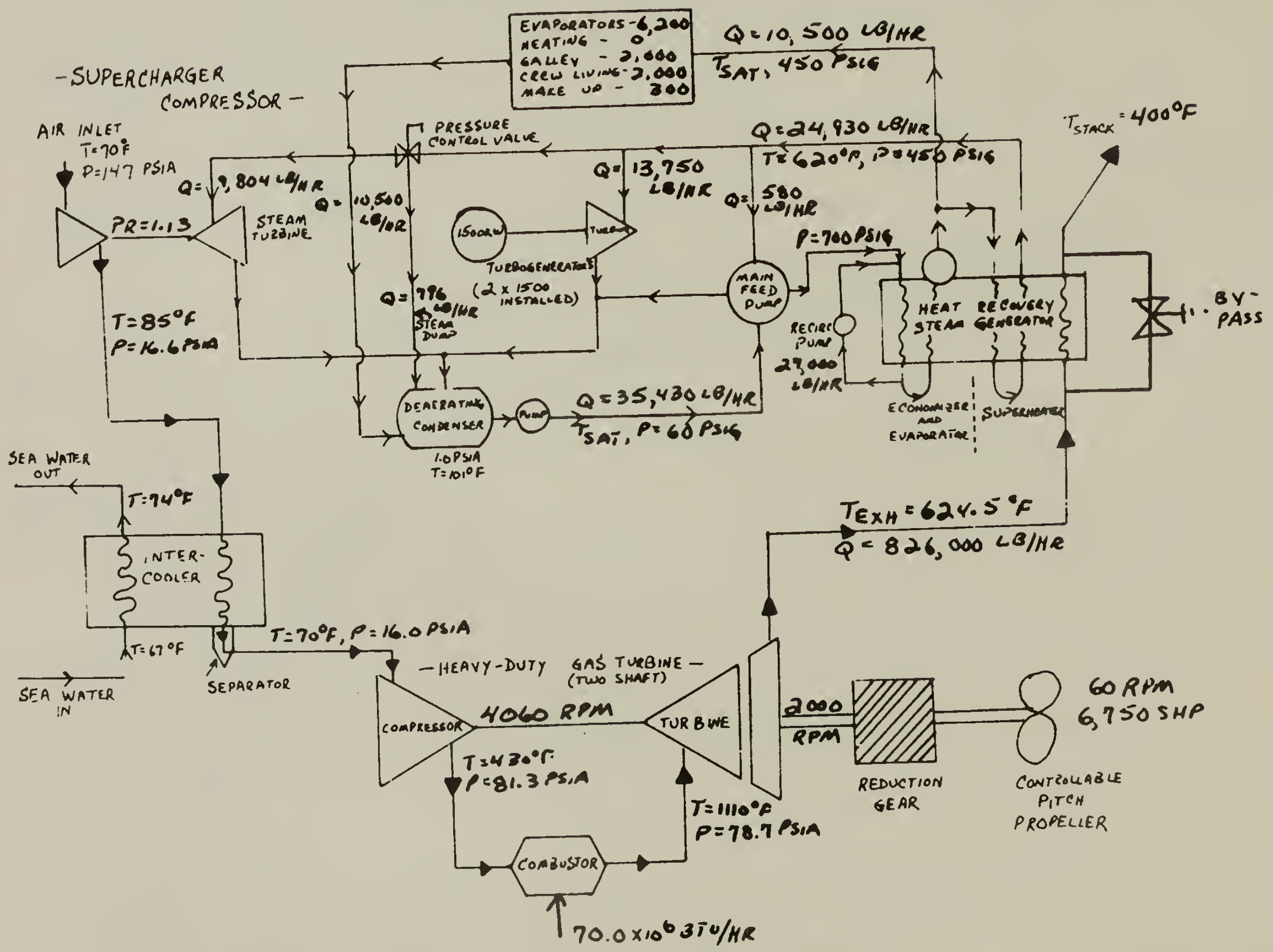
LEGEND

AOE-1 17 Knot Endurance
Conditions:
2 x 10,500 SHP
84 RPM
Design fuel rate .685
1140 KW Electric Ld.

SUPERCHARGED PLANT AT
17 KNOTS:

2 x 11,100 SHP
84 RPM
Supercharger SHP 2310
1140 KW Electric Ld.
(2 TG sets running)
Combined cycle overall
thermal efficiency:
0.258
All purpose design
fuel rate: $0.545 \frac{\text{lb}}{\text{SHP-hr}}$
Navy Distillate fuel
LHV: 18,100
Fuel/air ratio:1.013

Figure 3.24 Supercharged Power Plant Setup--17 Knots



SUPERCHARGED GAS TURBINE
AND STEAM PLANT

- 12 KNOTS -

LEGEND

AOE-1 12 Knot Replenishment conditions:
2 x 6750 SHP
60 RPM
Design fuel rate .750
3230 KW Electric Ld.

SUPERCHARGED PLANT AT
12 KNOTS:

2 x 6750 SHP
60 RPM
Supercharger SHP 1380
3230 KW Electric Ld.
(2 TG sets Running & 1-1000KW diesel)
Combined cycle overall thermal efficiency: 0.234
All prupose design fuel rate: 0.600 $\frac{\text{lb}}{\text{SHP-hr}}$
Navy Distillate fuel
LHV: 18,100
Fuel/air ratio: 1.010

Figure 3.25 Supercharged Power Plant Setup--12 Knots

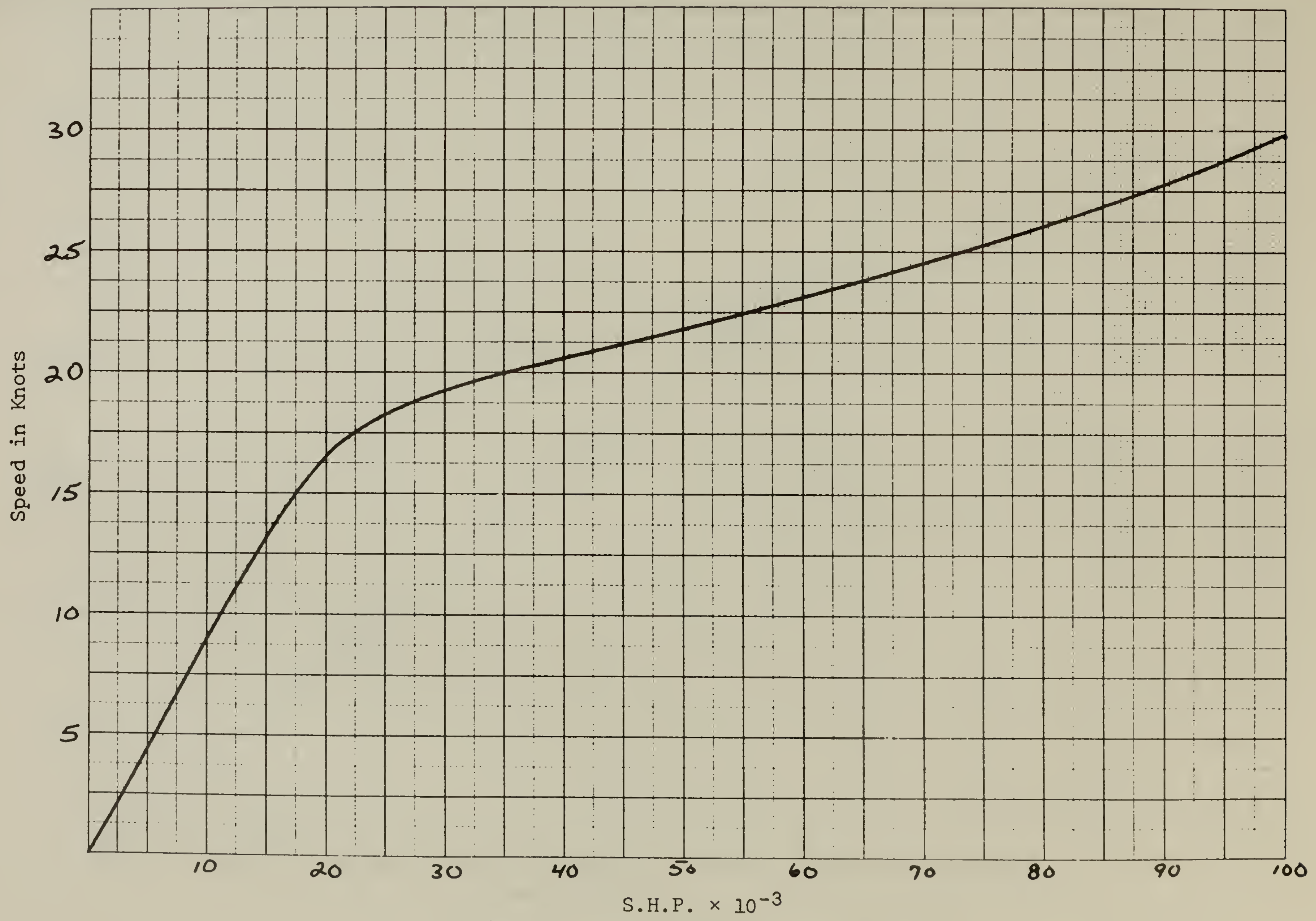


Figure 3.26 Total Shaft Horsepower versus Ship Speed

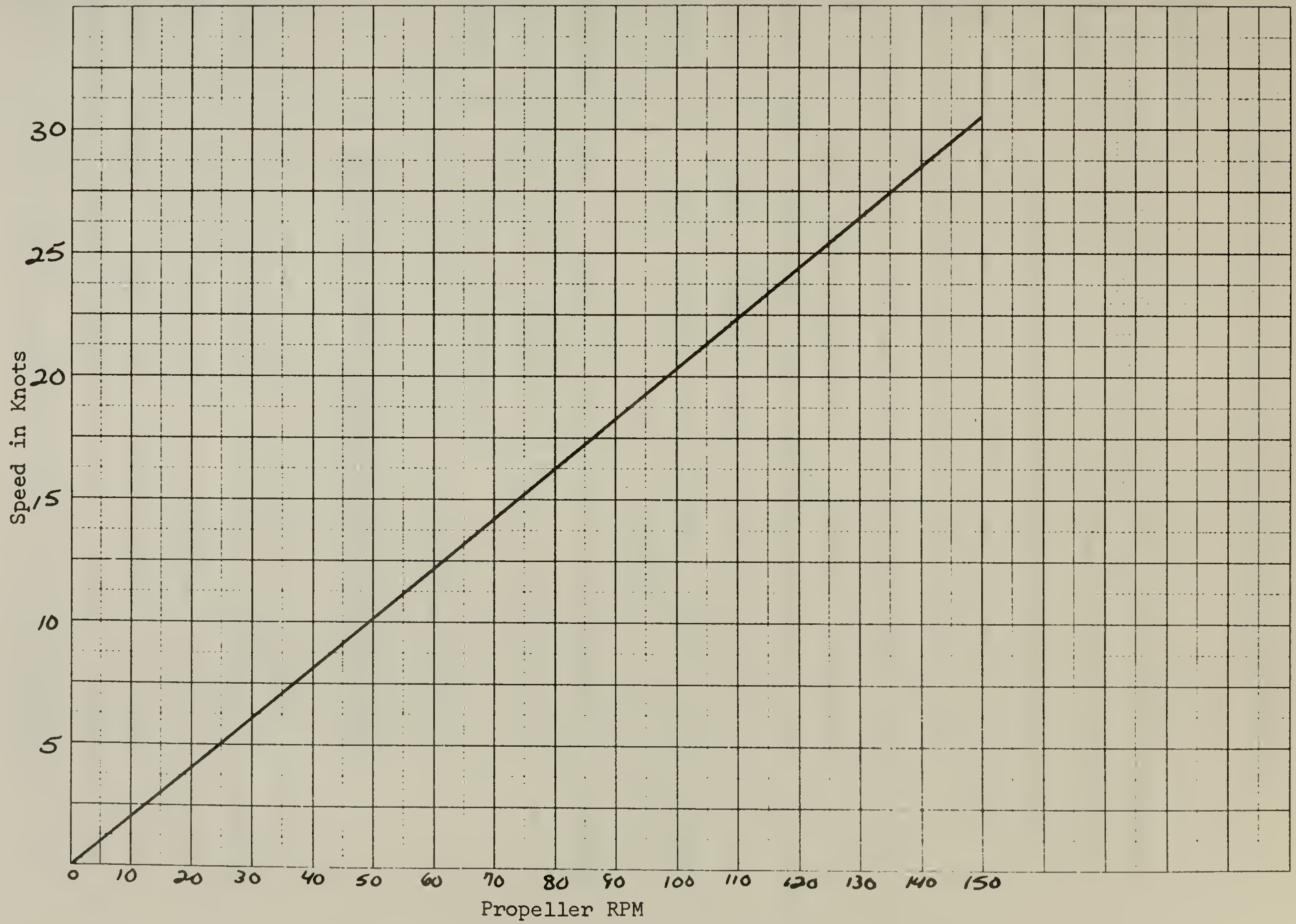


Figure 3.27 Shaft RPM versus Ship Speed

4. RELIABILITY, AVAILABILITY AND MAINTENANCE ANALYSIS

Reliability

The ability of any plant to function when required often separates the good plant on paper from the truly outstanding means of propulsion. While there is much disagreement among experts within the power industries concerning the best method to define and approach the problem, there are some generally accepted concepts with which to work.

Reliability has always been of paramount importance in ships, particularly in a nation's warships, where the national interest makes reliable systems a must. Offsetting this demand for greater and greater reliability is the technological complexity which has found its way into marine propulsion plants, just like other modern systems. Thus the marine designer must always be cautious not to compromise reliability. The supercharged gas turbine plant described in Chapter Three is an example of the integration of components into a relatively complex propulsion system. How does it stand up in the area of reliability?

Reliability is generally defined as the following ratio:

$$R= \frac{\text{Installed hours--(scheduled \& unscheduled outage)}}{\text{Installed hours}}$$

This reliability is called running reliability, or operating reliability, as opposed to starting reliability. Starting reliabilities are generally lower than running reliabilities, but for purposes of analysis, we shall consider operating reliabilities. Parenthetically, it is noted that the General Electric Model Series 5000 has had a starting reliability of 96.6% measured at random land-based machines during the last four years. (52)

The reliability formula used in this analysis is: (53)

$R = e^{-\lambda t}$ where: e = 2.71828, a constant

λ = Failures per hour of running time, also the reciprocal of Mean Time Between Failures, 1/MTBF

t = Period of time for the voyage run at a constant level of power.

Reliability of items combined in series is such that all components must work satisfactorily in order to obtain the required performance. The combined reliability for a series of three components is:

$$R = e^{-(\lambda_1 + \lambda_2 + \lambda_3)t}$$

With this method, the values of λ are shown in Table 4.1 and added. A more detailed analysis would consider redundant systems (e.g., two pumps in parallel) but for gross calculations, all component failure rates are considered additive:

	λ, Failure rate × 10^6 hours (53)(54)
Model Series 5000 Gas Turbine	69.4
Heat Recovery Steam Generator	15.0
Supercharger Steam Turbine & Compressor	11.0
Intercooler	3.0
Reduction Gear	0.7
Main Shaft and Bearings	0.5
Controllable and Reversible Pitch Prop.	16.0
Deaerating Main Condenser	4.0
Ship's Service Electrical System	21.5
Main Feed Pump	8.0
Main Condensate Pump	6.0
Main Condenser Vacuum Pumps	6.0
Main Lube Oil Cooler	2.0
Main Circulating Water Pump	4.0
	167.1

TABLE 4.1 Failure Rates of Major Components

For the purpose of this calculation, a ten-day voyage was chosen, i.e., 240 hours. Using this number, the reliability of the system is calculated:

$$R_{10} = e^{-(167.1 \times 10^{-6})(2.40 \times 10^{2})} = e^{-.0402} = 0.9608$$

For a twenty-day period, the reliability of the system is calculated:

$$R_{20} = e^{-(167.1 \times 10^{-6})(4.80 \times 10^{2})} = e^{-.0804} = 0.9237$$

The failure rate for the gas turbine comes from land-based data, and a correction is therefore made for the combined cycle in the marine environment by reducing the reliability figures by 1% in each case. The reliabilities thus become: (See Figure 4.1)

10-day voyage: R = 0.9512

20-day voyage: R = 0.9145

Availability

Availability is essentially a measure of the total time a plant is out of operation, both for scheduled and unscheduled maintenance. It is therefore defined as the following ratio:

$$A = \frac{\text{Installed hours--(scheduled \& unscheduled maintenance)}}{\text{Installed hours}}$$

The following figures represent approximate values for forced and scheduled outages for the Frame 5 gas turbine and steam plant.

	Scheduled Maintenance (52)(54)	Unscheduled Maintenance (Forced outage) (52)(54)
GAS TURBINE SYSTEM (Major, minor and service inspections)	148 hours	157 hours
STEAM CYCLE	40 hours	20 hours
TOTAL HOURS:	188 hours	177 hours

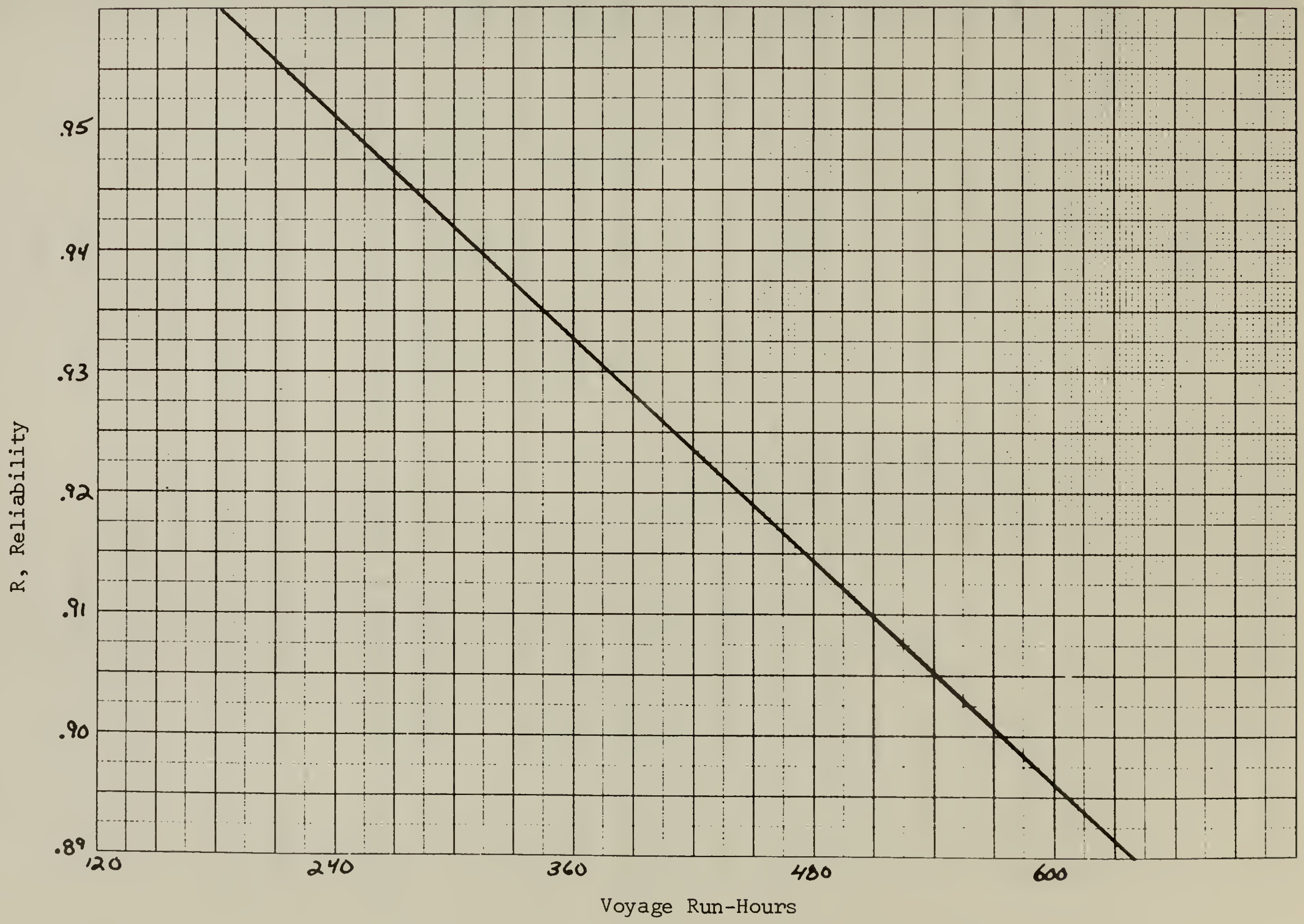


Figure 4.1 Supercharged Power Plant Reliability

The availability, then becomes (for installed hours equal to one year):

$$A = \frac{8760 - (188 + 177)}{8760} = 95.6\%$$

Maintenance

Gas Turbine System--The combustion gas turbine, as with any rotating power turbomachine requires a program of planned periodic inspections and replacement of parts in order to maintain operational reliability. The major structural components of the heavy-duty gas turbine are designed from long-established concepts derived from steam turbine technology. Along with the maintenance of the gas turbine itself, fuel and lube oil systems, control devices and other auxiliaries (such as the starting motor) require servicing from time to time. All of these functions may be performed in place--similar to the steam and diesel engine. This is in contrast to the replacement philosophy in which the entire unit is removed and returned to the factory for repair. With in-place repair, minor repairs and replacements are easily performed which would otherwise require complete removal and substitution.

It is worthwhile to review at this time the basic features of the heavy-duty gas turbine which make it so readily accessible and maintainable. They are: (55)

1) All casings, both turbine and compressor, are split on the horizontal centerline and may be lifted individually for access to all internal parts.

2) With the upper half of the compressor casing removed, all stator blades can be slid circumferentially out of the casings for inspection or replacement. This can be readily accomplished even with the rotor in place.

3. With the upper half of the turbine shell lifted, both halves of the first-stage nozzle assembly may be removed for inspection or repair, with the rotor in place. The interstage diffuser and diaphragm may also be removed by first removing the radial locating pins. The variable-angle second-stage nozzle can then be removed radially inward, after the actuating levers and inner side walls have been removed.
4. All turbine buckets are moment weighed and charted for assembly so that, in an emergency, they can be replaced without rebalancing the rotor. Normally, the rotor balance should be checked after replacing any parts.
5. All bearing housings and liners are split on the horizontal centerline so that they may be inspected and replaced when necessary. The lower half of the liners may be removed with the rotor in place.
6. All seals and shaft packings are separate from the main housings and may be readily removed and replaced.
7. Fuel nozzles and combustion liners can be removed for inspection, maintenance, or replacement without lifting or moving any of the casings. Removable cover plates close the ends of each combustion chamber.
8. All major accessories, as well as filters and coolers, are separate assemblies that are readily accessible for inspection or maintenance. They may also be individually replaced if necessary.

There are four major inspection schedules which are recommended for the industrial gas turbine. (56) They are:

a) Running inspections
b) Service inspections
c) Minor inspections
d) Major inspections and overhaul

The running inspection, as the name implies, is the day-to-day observation of temperatures, pressures and other indications which go to make up the critical check points for the operation of a high-speed turbomachine. These tasks would be performed by ship's force while on watch, under the supervision of the Engineering Officer of the Watch (see Appendix F).

The service inspection consists of a short shutdown period to inspect combustion liners and fuel nozzles. A fouled combustion system can result in much shortened life of the downstream turbine nozzles and blading. This inspection is done at approximately 6000-hour intervals. See Figure 4.2 (57).

A minor inspection is carried out at the 12,000-hour point. This is basically a hot-gas inspection and includes a visual inspection of the turbine nozzles and buckets. The top half of the turbine shell must be removed to accomplish this. See Figure 4.3 (58).

The major inspection or overhaul is normally at the 24,000-hour mark. This includes all inspections done in the service and minor inspections plus a look at the compressor blading and all bearings. See Figure 4.4 (59). See also Table 4.2 (24).

Recommended spares are listed in Table 4.3 (60).

Steam Generation System--The major item of concern here is the heat recovery steam generator. The waste heat boiler gas sides are prone to fouling

CUMULATIVE HOURS OPERATION	ESTIMATED MAN HOURS	DESCRIPTION
4000	6	REMOVE FUEL NOZZLES FOR CLEANING ABOARD SHIP.
6000	6	INSPECT COMBUSTION LINERS, TRANSITION PIECES, AND BORE-SCOPE INSPECTION OF 1ST STAGE NOZZLES.
8000	6	REMOVE FUEL NOZZLES FOR CLEANING ABOARD SHIP.

REPEAT AFTER MINOR INSPECTION

Figure 4.2 Estimated Service Inspection Schedule (57)

CUMULATIVE HOURS OPERATION	ESTIMATED MAN HOURS	DESCRIPTION
12,000	24	REPLACE COMBUSTION LINERS AND TRANSITION PIECES. (REPAIR OLD PIECES FOR USE AS SPARES).
	64	REPLACE 1ST STAGE NOZZLES. (REPAIR OLD NOZZLES FOR USE AS SPARES).
	1	REMOVE F.O. PUMP FOR REPAIR.
	1	REMOVE FLOW DIVIDER FOR REPAIR.
	6	REMOVE FUEL NOZZLES FOR CLEANING ABOARD SHIP

REPEAT AFTER MAJOR INSPECTION

Figure 4.3 Estimated Minor Inspection Schedule (58)

CUMULATIVE HOURS OPERATION	ESTIMATED MAN HOURS	DESCRIPTION
24,000	6	REMOVE FUEL NOZZLES FOR CLEANING ABOARD SHIP.
	24	REPLACE COMBUSTION LINERS AND TRANSITION PIECES. (REPAIR OLD PIECES FOR USE AS SPARES).
	128	REPLACE 1ST AND 2ND STAGE NOZZLES. (REPAIR OLD NOZZLES FOR USE AS SPARES).
	300	INSPECT 1ST STAGE BUCKETS, REPAIR AS NECESSARY. LIFT COMPRESSOR CASINGS AND INSPECT BLADING. INSPECT BEARINGS.

REPEAT AT 24,000 HOUR INTERVALS

Figure 4.4 Estimated Major Inspection Schedule (59)

TABLE 4.2 Estimate of Maintenance Schedule (24)

Cumulative Hours Operation	Man Hours Required	Work Performed by	Description
4,000	6	Crew	Remove F.O. nozzles for repair aboard ship
6,000	6	Crew	Inspect combustion liners, transition pieces and 1st-stage nozzles
8,000	6	Crew	Remove F.O. nozzles for repair aboard ship
12,000	24	Supervised Crew	Replace combustion liners and transition pieces
	64	Supervised Crew	Replace first-stage nozzles
	1	Crew	Remove F.O. pump for repair
	1	Crew	Remove flow divider for repair
	6	Crew	Remove F.O. nozzles for repair aboard ship
16,000	6	Crew	Remove F.O. nozzles for repair aboard ship
18,000	6	Crew	Inspect combustion liners, transition pieces and 1st-stage nozzles
20,000	6	Crew	Remove F.O. nozzles for repair aboard ship
24,000	6	Crew	Remove F.O. nozzles for repair aboard ship
	24	Supervised Crew	Replace combustion liners and transition pieces
	128	Supervised Crew	Replace 1st-stage and 2nd-stage nozzles
	300	Supervised Crew	Inspect 1st-stage buckets; repair as necessary

Cumulative Hours Operation	Man Hours Required	Work Performed by	Description
28,000	6	Crew	Remove F.O. nozzles for repair aboard ship
30,000	6	Crew	Inspect combustion liners, transition pieces and 1st-stage nozzles
32,000	6	Crew	Remove F.O. nozzles for repair aboard ship
36,000	24	Supervised Crew	Replace combustion liners and transition pieces
	64	Supervised Crew	Replace first-stage nozzles
	1	Crew	Remove F.O. pump for repair
	1	Crew	Remove flow divider for repair
	6	Crew	Remove F.O. nozzles for repair aboard ship
40,000	6	Crew	Remove F. O. nozzles for repair aboard ship
42,000	6	Crew	Inspect combustion liners, transition pieces and 1st-stage nozzles
44,000	6	Crew	Remove F.O. nozzles for repair aboard ship
48,000	6	Crew	Remove F.O. nozzles for repair aboard ship
	24	Supervised Crew	Replace combustion liners and transition pieces
	128	Supervised Crew	Replace 1st-stage and 2nd-stage nozzles
	300	Supervised Crew	Inspect first-stage buckets; repair as necessary

(TABLE 4.2)

TABLE 4.3 RECOMMENDED SHIPS SPARES REQUIREMENTS PER ENGINE

MARINE TWO-SHAFT GAS TURBINES--(FIRST 24,000-HOUR OVERHAUL) (60)

PARTS LIST	NO. SPARES
COMBUSTION	
Cap & liner	12
Crossfire tubes	12
Sight glass	12
Gaskets	12
Retaining rings	80
Sealer SP	12
Transition piece	2
Fuel nozzle	12
Spark plugs	12
Gaskets (plugs)	12
Cores	2
FUEL SYSTEM	
Fuel oil stop valve	1
Packing	2
Klozure gasket	1
Rings	2
Valve seat	1
Valve disc	1
Oil seal	1
Gasket	1
Connector tubing	12
Check valve	12
Unions	36
Fuel pump	1
Flow divider	1
Ignition transformer	2
Flame detector	2
Spark plug leads	2
Filters (duplex)	2
Gaskets (system)	15
BEARINGS	
#1 Bearing liner	1
Deflector	1
Thrust shoes	2 sets
#2 Bearing liner	1
Deflector	2
#3 Bearing liner	1
Deflector	2
BEARINGS (Cont'd.)	
#4 Bearing liner	1
Deflector	1
Air deflectors	2
LUBE OIL SYSTEM	
Main lube oil pump	1
Pump gear	2
Bearings	4
Retaining rings	2
Bearing feed/drain gaskets	15
Auxiliary lube oil pump	1
Impeller	1
Ball bearings	3
Coupling	1
Coupling bolts	6
Bearing seals	2
Oil tanks internal gaskets	15
Oil tanks external "	10
Filters, (duplex) main	2
Control oil filters	1
Control oil gaskets	2
COUPLINGS	
Load gear	1
Accessory drive	1
Starting gear	1
Coupling bolts	8
Coupling nuts	8
COOLING SYSTEM	
Water pump	1
Seals	2
Bearings	2
Gaskets (system)	50
Impeller	1
CONTROL SYSTEM	
"SPEEDTRONIC" control cards	250

TABLE 4.3 (Continued)

PARTS LIST	NO. SPARES
CONTROL SYSTEM (Cont'd.)	
Fuses	15
Resistors	50
Lamps (indicator)	30
Control switch type SBM	10
Thermocouples (exhaust control)	12
Exhaust trip	6
Turbine Int. wheel space	2
Turbine shell	1
Nozzle	4
TOOLS	
"SPEEDTRONIC" calibrator	1
Frahm tachometer	1
Vibration indicator (light beam)	1
Volt/amp meter hook-on	1
Borescope	1
Card puller	1

depending on which type of fuel is burned. Soot blowers are installed in the boiler to periodically clear away soot and other products of combustion which tend to reduce the life and efficiency of the boiler. Waterside chemistry for the boiler is also of utmost importance as the successful operation of steam generating equipment depends upon a rigid control of feed water to assure freedom from scale formation and corrosion of steam contacted surfaces. The three most critical items to be controlled are hardness (0 ppm), oxygen (.007 ppm), and alkalinity (8.0 p.H). Boiler water hardness is generally controlled with combinations of trisodium phosphate and other sodium and phosphate compounds. These form a sludge which may be removed from the boiler by frequent blowdowns. The oxygen is kept under control through the deaerating condenser (no deareating feed tank), and by the addition of sodium sulfite. The latter method of control is commonly used in submarine power plants. (61)

Maintenance aspects of the other portions of the steam cycle consist of normal routine maintenance as with any steam plant. Steam turbines are essentially maintenance free--thus the bulk of the maintenance requirements would revert to pump repacking, reseating valves, rewinding motors and looking for water leakage. The relatively low pressure of 450 psig should make the plant relatively easy to maintain.

Maintenance Costs

It is estimated that most of the work on the cycle will be able to be accomplished by a well-trained ship's force. Some technical supervision will no doubt be required to assist from time to time with the gas turbine, and this need is reflected in the estimated maintenance costs.

		TOTALS:
GAS TURBINE (52)(54)		
Annual charge & technical assistance	$10,500	
Parts and repair @ 5.40 per fired hour × two plants for 5250 running hours	56,600	
	$67,100	$67,100
STEAM SYSTEM: (54)		
Technical assistance	$ 2,500	
Parts and repair	37,500	
	$40,000	$40,000
		$107,100

5. LIFE CYCLE COSTS COMPARED WITH PRESENTLY-INSTALLED STEAM PLANT

The costing of power plants on a comparative basis is not an easy task, as cost data varies from reference to reference and thus any analysis tends to be a subjective one. While there are many sources, the primary one was the General Electric Company estimates, (54)(60) tempered with other economic studies. The basic philosophy was to approach costs on the high side, as much of the data is more than one year old. It should be noted that detailed costs are made only for the supercharged cycle contained in this thesis--the only attempt to make a comparison with the presently-installed steam plant is in the summary of total cost data in Table 5.1. The costing of the conventional steam plant is admittedly a gross estimate, and was made only to see whether the supercharged plant is within the same range. Additionally, one might like to ask the question, "If the Navy today decided to build an additional Fast Combat Support Ship, how much might a gas turbine power plant cost as compared with a steam plant?"

The results of Table 5.1 show, not surprisingly, that the total of present worth of annual costs plus acquisition costs for the gas turbine plants exceeds that of the steam plant. The most significant figures are the fuel costs which are close to 30% higher for the steam plant. While the Navy historically has seemed to feel fuel is a small part of their operating costs, five of these ships operating for twenty years represents a sizable amount of money. Perhaps the Navy, in the corporate sense, should take a closer look at annual fuel costs.

TABLE 5.1 Summary of Cost Data

	SUPERCHARGED CYCLE	PRESENTLY INSTALLED STEAM PLANT (GROSS EST.)
ACQUISITION COSTS	$19,557,040	$11,700,000
OPERATIONAL COSTS		
Annual Fuel	$1,804,000	$2,419,000
Annual Maintenance	117,810	82,000
	$1,921,810	$2,501,000
NET PRESENT VALUE of operating costs for 20 years @ 12% interest:	$14,250,000	$18,700,000
	$33,807,040	$30,400,000

TABLE 5.2 Summary of Acquisition Costs for the Supercharged Cycle

MATERIAL:		
a) Main propulsion, gas turbine and steam plant, per shaft	$ 5,820,800 × 2 shafts =	$ 11,641,600
b) Electrical	862,000 × 2 shafts =	1,724,000
c) Miscellaneous	236,000 × 2 shafts =	472,000
		$ 13,837,600
INSTALLATION:	$ 1,834,400 × 2 shafts =	$ 3,668,800
		$ 17,506,400
OVERHEAD AND ADMINISTRATIVE EXPENSES @ 10% =		$ 1,750,640
		$ 19,257,040
TRIALS:		$ 300,000
		$ 19,557,040

TABLE 5.3 Detailed Breakdown of Acquisition and Installation Costs

(Per Engine Room)

(Detailed list of equipment may be found in Appendix E)

		ACQUISITION (54)(60)	INSTALLATION (54)
MAIN PROPULSION			
a.	General Electric MS 5322B gas turbine, inlethouse & demister, Speedtronic controls, propulsion control bridge remote, dual L.O. coolers & standard accessories	$ 2,020,000	$ 95,000
b.	Reduction Gear	860,000	60,000
c.	Supercharger turbine & compressor	788,000	10,000
d.	Heat recovery steam generator	500,000	95,000
e.	Auxiliary boiler	40,000	15,000
f.	Ductwork	91,000	210,000
g.	Intercooler	120,000	11,000
h.	Deaerating condenser	140,000	9,600
i.	Vacuum pumps	16,800	2,000
j.	Salt water circulating pump	40,000	4,000
k.	Controllable pitch propeller and shafting	840,000	77,000
l.	Piping	162,000	780,000
m.	Main feed pumps	42,000	4,800
n.	Condensate pumps	25,000	3,000
o.	Evaporators	120,000	10,000
p.	Air compressors	36,000	1,500
	SUBTOTALS:	$ 5,820,800	$ 1,387,900

TABLE 5.3 (Continued)

	ACQUISITION	INSTALLATION
ELECTRICAL		
a. Turbo-generators	$ 512,000	$ 19,000
b. Diesel generator	125,000	10,000
c. Wireways	65,000	96,000
d. Switchboards	160,000	20,000
SUBTOTALS:	$ 862,000	$ 145,000
MISCELLANEOUS		
a. Engine Room outfit	$ 186,000	$ 180,000
b. Drawings	6,000	48,000
c. Automation (steam side only for acquisition)	8,000	72,000
d. Shaft brake	36,000	1,500
SUBTOTALS:	$ 236,000	$ 301,500

TOTALS: Found in Table 5.2

OPERATING COSTS

a. Annual Fuel Costs--based on engine running 5,250 hours per year.

These hours are arbitrarily broken down into the following schedule:

i. Assume 40% of time at full power = 2,100 hours

ii. Assume 20% of time at 17 knots = 1,050 hours

iii. Assume 40% of time at 12 knots = 2,100 hours

i. Full Power: Navy distillate fuel (62)

$$2100 \text{ hours} \times \frac{\$4.35}{\text{BBL}} \times \frac{1 \text{ BBL}}{305 \text{ lb}} \times \frac{.444 \text{ lb}}{\text{SHP-HR}} \times 100{,}000 \text{ SHP} = \$\ 1{,}390{,}000$$

TABLE 5.3 (Continued)

ii. 17 knot endurance: Navy distillate fuel

$$1050 \text{ hours} \times \frac{\$4.35}{\text{BBL}} \times \frac{1 \text{ BBL}}{305 \text{ lb}} \times \frac{.545 \text{ lb}}{\text{SHP-HR}} \times 21{,}000 \text{ SHP} = \$\ 171{,}500$$

iii. 12 knot replenishment: Navy distillate fuel

$$2100 \text{ hours} \times \frac{\$4.35}{\text{BBL}} \times \frac{1 \text{ BBL}}{305 \text{ lb}} \times \frac{.60 \text{ lb}}{\text{SHP-HR}} \times 13{,}500 \text{ SHP} = \$\ 243{,}000$$

TOTAL: $ 1,804,000

ANNUAL MAINTENANCE COSTS

From Chapter 4, annual cost is estimated to be \$107,100. Since approximately half of the maintenance cost comes from land-based gas turbine data, (as opposed to those operating in a constant salt air environment) 1% is added to the cost figure. While this is admittedly arbitrary, it is highly unlikely that the costs will go down in the salt air environment, and thus a conservative estimate would put the figure higher. Maintenance costs are rarely overestimated. This figure then becomes:

\$ 117,810

LIFE CYCLE COSTS

Using net present value concepts which relate future expenditures to current dollars, the total of fuel and maintenance costs are computed. An annual rate of 12% interest is assumed and time is for a 20-year period. The basic relationship here is: (63)

Where i = interest rate, N = time in years, P = net present value, and R = annual cost of fuel and maintenance,

$$P = R\left[\frac{(1+i)^N - 1}{i(1+i)^N}\right]$$

The life cycle cost thus becomes \$14,250,000.

ESTIMATED FUEL COSTS OF PRESENTLY-INSTALLED STEAM PLANT (51)

i. Full Power: Navy distillate fuel oil

$$2100 \text{ hours} \times \frac{\$4.35}{\text{BBL}} \times \frac{1 \text{ BBL}}{305 \text{ lb}} \times \frac{.632 \text{ lb}}{\text{SHP-HR}} \times 100{,}000 \text{ SHP} = \$\ 1{,}900{,}000$$

ii. 17 knots: Navy distillate fuel oil

$$1050 \text{ hours} \times \frac{\$4.35}{\text{BBL}} \times \frac{1 \text{ BBL}}{305 \text{ lb}} \times \frac{.685 \text{ lb}}{\text{SHP-HR}} \times 21{,}000 \text{ SHP} = \$\ 216{,}000$$

iii. 12 knots: Navy distillate fuel oil

$$2100 \text{ hours} \times \frac{\$4.35}{\text{BBL}} \times \frac{1 \text{ BBL}}{305 \text{ lb}} \times \frac{.750 \text{ lb}}{\text{SHP-HR}} \times 13{,}500 \text{ SHP} = \$ \quad 303{,}000$$

$ 2,419,000

6. CONCLUSIONS AND RECOMMENDATIONS

The main conclusions of this feasibility study in using a supercharged heavy-duty gas turbine and steam cycle for marine propulsion are as follows:

1) It would be feasible to place a large heavy-duty supercharged gas turbine and steam cycle aboard a large Naval Auxiliary ship such as the Sacramento Class (AOE-1). The plant will fit within the presently installed engine rooms with no significant weight increase. Some modification to the hull superstructure would be required in order to provide for increased air flow to the gas turbine.

2) The specific fuel rate at full power, 0.444, is significantly better than that of the presently installed steam plant. The SFC is also better at lower power ranges, and the annual saving in fuel costs to the Navy (with both plants burning Navy Distillate fuel) would be on the order of 30%, with longer endurance capability.

3) The excellent thermal efficiency and fuel rates are comparable with those of the aircraft derivative gas turbine. At lower power levels, the ability to use the waste heat from the gas turbines enables the heavy-duty gas turbine to maintain a "flatter" fuel rate curve. It appears that with the performance characteristics of the heavy-duty gas turbine as indicated by this analysis, more consideration to their use should be given by the Navy for large displacement ships.

4) While supercharging of the gas turbine presents an interesting thermodynamic consideration, it is concluded that steam produced in a waste heat boiler can best be utilized by coupling a steam turbine directly to the shaft, as in the case of the STAG cycle (Figure 2.3), which has a good specific fuel rate.

5) The life cycle cost of a steam plant for the AOE-1 class would be less than that of the supercharged gas turbine cycle, primarily due to initial acquisition costs.

6) The control of a supercharged cycle is a complex one and careful consideration must be given to off-design points, i.e., steam pressure and steam flow changes as the gas turbine changes power levels, specifically, exhaust gas temperature and flow.

7) The analysis showed that the supercharged cycle, as presently configured, produced unacceptable levels of 'dumped' steam at high power levels. At lower power levels, e.g., while the ship was replenishing at 12 knots (large KW load alongside), there is insufficient steam to generate electrical power without using the emergency diesel and without reducing hotel load. This must be considered a totally unacceptable condition while alongside another ship.

8) The use of an intercooler aboard ship is worthy of consideration in a gas turbine cycle, as there is a large heat sink (the ocean) available which is normally not used. Intercooling at temperature levels considered for this plant would not normally be recommended. Intercooling would be more beneficial to the thermal efficiency of the cycle if accomplished at higher temperatures.

However, intercooling in a marine environment is particularly desirable as there will be a significant reduction in airborne sea salts. Sodium salt reduction reduces hot part corrosion of the gas turbine.

9) The AOE-1 Class of auxiliary ship would be able to steam in excess of 24 knots with the steam plant completely secured, i.e., on gas turbine power alone. Electrical take-home power could be supplied by the emergency diesel generator.

The principle recommendation for further work in the area of using heavy-duty gas turbines for ship propulsion in the Navy has already been alluded to in the conclusions--that is, a closer look at the possibilities for their use is warranted. The industrial gas turbine has been used in industry for many years. While only one ship was ever propelled by a U.S. heavy-duty gas turbine (SS John Sergeant), the General Electric Company alone has 22 units in marine environments with operating time in excess of 1,000,000 hours. While this in itself does not show that they will perform perfectly aboard a moving vehicle, the durability, ruggedness and in-place maintenance of the machines, coupled with exceptional reliability and availability figures makes them a desirable option in marine power plants. Maintenance costs and acquisition costs remain an unknown an at present tend to be on the high side, and these are definite disadvantages.

A possible modification to the combined cycle for the industrial gas turbine would be the regenerative cycle. As stated in Chapter 2, however, a regenerator should be used with an intercooler and these two items make a very large size and weight factor in propulsion system design. Additionally, regenerators may increase maintenance costs considerably. Thus

it is recommended that an industrial gas turbine be used with a waste heat boiler at low steam pressures. This allows power generation and other auxiliaries as well as hotel services, which would have to be provided by a gas turbine or diesel generator, and an auxiliary boiler. The main advantage in the combined cycle is simply one of economics--why exhaust gases to the atmosphere at 700° or 800° F. when they can be used to do work? To those who feel that combined plants are the "worst of both worlds," it must be recalled that much sentiment toward steam within and outside the Navy has been colored by 1200 psig refractory boilers designed as long ago as 15 years. Thus, it is submitted that the steam portion of this cycle (particularly with an unfired HRSG) would not in fact be a maintenance nightmare. With respect to the firing versus unfired waste heat recovery steam generator, the boiler could very easily be fired to prevent the low steam flows when at low power. Additionally, this allows a mode of auxiliary power when the propulsion gas turbine is not on the line and could be used for start up in lieu of the gas turbine starting system described in Appendix C. The question of forced and natural circulation is nearly a straight trade off--the natural circulation boiler is easier to clean and the watersides tend to remain cleaner, while the forced circulation is smaller and saves on size and weight. The forced circulation boiler was chosen for the latter reason and also due to the fact that economizer water was recirculated to bring feed water temperature up to an acceptable level.

The steam cycle in a combined cycle should be kept as simple as possible--thus the decision was made to go with a deaerating condenser. It is recommended that this concept be used for shipboard combined cycle plants,

particularly at low pressures and temperatures, where dissolved oxygen and pH requirements are somewhat eased. With respect to pressure levels for the steam plant, again a trade-off must be made. Lower steam pressures mean lower stack temperatures and a more efficient plant--they do, however, tend to increase the size of steam machinery and piping.

One consideration for power generation is the attachment of a ships service generator to the main shaft--this design is being implemented on the ships for Broken Hill Proprietary, Co., Ltd., previously mentioned. The method used there is to maintain frequency within 10% by placing larger emphasis on changing propeller pitch, that is, propeller pitch varies to hold essentially constant speed for the attached generators. Additionally, it would be possible to run an attached generator with variable speed yet constant frequency through an electronic package.

Another possible recommendation in the design of a combined cycle plant would be to use gas turbine generators. They would be particularly useful during warmup and starting time, as they could exhaust directly to the waste heat recovery steam generator and commence producing steam, which in the case of the supercharged cycle, would allow the supercharger to start.

The original design for this supercharged cycle included the use of steam injection directly into the combustion chamber. This reduces nitrous oxide levels significantly and would be desirable when entering or leaving port.(64) Additionally, power is increased with the increased mass flow and for each 1% of steam injection (related to air flow) approximately a 4% increase in power is realized. The integration of this component into the system was discarded as insufficient steam flow was available

at low power levels. Additionally, trade-offs between doubling the evaporator capacity (to 120,000 GPD) with the added cost, as opposed to the actual net gain in power and pollution prevention made it infeasible in a shipboard environment. Steam fuel atomization and water injection are also possible alternatives, but these, like steam injection, are probably best utilized in a land-based power plant.

A final recommendation would be to put another loop into the steam control system--an input which would regulate the waste heat boiler bypass damper valve if steam pressure began to rise uncontrollably and safety of plant and personnel were endangered.

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APPENDIX A.--THERMODYNAMIC CONSIDERATIONS OF THE CYCLE

A.1 Full Power Calculations (Fig 3.1)

Figure A.1 shows the T-S diagram (qualitatively) for the supercharged gas turbine plant. Assumptions used in the sample calculations:

G. E. MS 5322(B) GAS TURBINE

1.	Inlet temperature (to both supercharger and gas turbine):	70° F. = 530° R.
2.	Inlet Pressure to Supercharger:	14.7 psia
3.	Supercharger Pressure Ratio:	1.4:1
4.	G.T. Inlet Loss:	4" H_2O
5.	G.T. Outlet Loss:	10" H_2O
6.	G.T. Firing Temperature:	1700° F.
7.	Pressure Drop, Combustion Chambers:	3%
8.	G. T. Pressure Ratio:	8.2:1
9.	Pressure Drop, Supercharger Outlet to G. T. Inlet:	0.6 psia
10.	Supercharger Compressor Efficiency (Isentropic):	0.85
11.	G.T. Compressor Efficiency (Isentropic):	0.86
12.	G.T. High Pressure Turbine Efficiency (Polytropic):	0.87
13.	G. T. Low Pressure Turbine Efficiency (Polytropic):	0.84
14.	Base G.T. Flow Rate:	256.3 lbm/sec
15.	Injection Temperature	67° F.

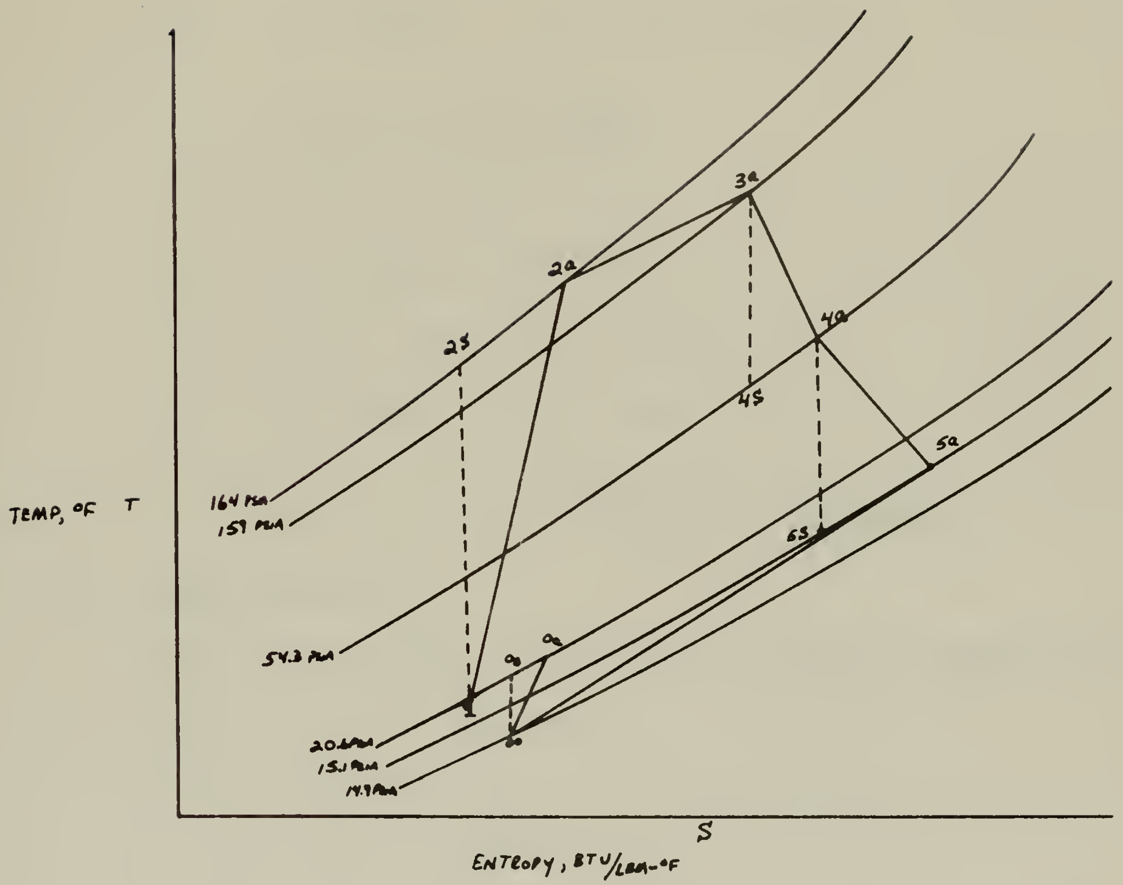


Figure A.1 T-S Diagram for Supercharged Power Plant

a.) SUPERCHARGER WORK (Using Reference 65)

@ $T_{00} = 530° \text{ R.}, P_{r_{00}} = 1.640 \quad h_{00} = 16.8$ Btu/lbm

$P_{00} = 14.7$ psia, $P_0 = 20.6 \text{ lbf/in}^2$

Where: s- Isentropic

a - Actual

$\therefore \quad P_{r_0} = 1.4\ (1.640) = 2.30$

@ $P_{r_0} \quad T_{0_s} = 548° \text{ R.}, \quad h_{0_s} = 29.760$ Btu/lbm

$$\eta_{sc} = .85 = \frac{h_{0_s} - h_{00}}{h_{0_a} - h_{00}}$$

$\therefore\ h_{0_a} = 32.1$ Btu/lbm

$T_{0_a} = 591.0° \text{ R.} = 131.2° \text{ F.}$

$\dot{m}_{sc} = 1.4\ (256.3) = 358.0$ lb/sec

$$W_{sc} = \dot{m}_{sc}\ (h_{0_a} - h_{00}) = \frac{(358)(15.3)}{.7074} = \underline{\underline{7{,}750 \text{ H.P.}}}$$

b.) GAS TURBINE WORK

Pressure drop through intercooler and ducting. Temperature reduced to 70° F. in intercooler.

$P_1 = P_0 - .6 = (20.6 - .6) = 20.0 \text{ lb/in}^2$

$T_1 = 70^{\circ} \text{ F.} = 530^{\circ} \text{ R.}$

@ $70° \text{ F.}, P_1 = 20 \text{ lbf/in}^2, h_1 = 16.8 \text{ Btu/lbm}, P_{r_1} = 1.640$

$P_2 = 8.2(20) = 164 \text{ lbf/in}^2$ and $P_{r_2} = 1.64\ (8.2) = 13.45$

@ $P_{r_2}, T_{2_s} = 500.3° \text{ F.}, h_{2_s} = 121.275$ Btu/lbm

$$\eta_c = .86 = \frac{h_{2_s} - h_1}{h_{2_a} - h_1} \quad \therefore \quad h_{2_a} = 138.28 \text{ Btu/lbm}$$

b.1) COMPRESSOR WORK

$$HP_c = \dot{m}_{GT} \Delta h = \dot{m}_{GT}(h_{2_a} - h_1) = \frac{(341.53)(121.48)}{.7074}$$

($\dot{m}_{GT}$ corrected for air temp. not @ 59°F.) = $\underline{\underline{58{,}650 \text{ HP}}}$

b.2) HIGH PRESSURE TURBINE (Δh same as Compressor Work)

$@\ T_{3_a} = 1700^\circ$ F. $= 2160^\circ$ R., $h_{3_a} = 439.92$, $P_{r_3} = 302.403$

$h_{4_a} = h_{3_a} - \Delta h_{HP} = 439.920 - 121.480 = 318.440$ Btu/lbm

$T_{4_a} = 1262^\circ$ F.

$$\eta_{HP} = .87 = \frac{h_{3a} - h_{4a}}{h_{3a} - h_{4s}}$$

$\therefore\ h_{4_s} = 300.28$ Btu/lbm

$@\ h_{4_s}$, $P_{r4_s} = 103.2$, $T_{4_s} = 1192.5^\circ$ F.

Pressure Ratio: $\frac{302.403}{103.2} = 2.93 \quad \therefore\ P_4 = \frac{159}{2.93} = 54.27$ lb/in^2

b.3) LOW PRESSURE TURBINE (Including 3% Combustor Loss)

Gases now expand to 10" back pressure, 15.1 lb^F/in^2

$@\ T_{4_a} = 1262^\circ$ F., $P_{r4_a} = 121.2$, $h_{4_a} = 318.440$

Pressure ratio $= \frac{54.27}{15.1} = 3.59 \quad \therefore\ P_{r5_s} = \frac{121.2}{3.59} = 33.66$

@ 33.66, $h_{5_s} = 189.920$ Btu/lbm $T_{5_s} = 773.5^\circ$ F.

$$\eta_{LP} = .84 = \frac{h_{4a} - h_{5a}}{h_{4a} - h_{5s}} \quad \therefore\ h_{5_a} = 210.77 \text{ Btu/lbm}$$

$T_{5_a} = 853^\circ$ F. (Exhaust temperature)

b.4) WORK OUT TO SHAFT

Assume: a) Loss in power correction from 70° F. to 59° F. is 0.96 (Figure C.2)

b) Fuel air ratio = 1.0185

c. Gear efficiency = 0.985

$$SHP = \dot{m}_{LP}\ \Delta h = \frac{(341.53)(107.95)(.96)(1.0185)(.985)}{.7074}$$

$$= \underline{\underline{50{,}194 HD}}$$

c.) STEAM CYCLE--See Table 3.6 and Reference (66)

Steam Conditions:

P = 450° psig, T = 750° F., 102,000 lb/hr

c.1) TURBOGENERATOR REQUIREMENTS (51)--4000 KW/Ship (Constant line pressure @ 450 psig); or 2000 KW/Engine Room (Two generators)

Theoretical Steam Rate:

$h_0 = 1389.3$

$h_p = \underline{921.0}$ @ 1 psia

468.3

$$TSR = \frac{3412.1}{h_0 - h_p}$$

$$TSR = \frac{3412.1}{468.3} = 7.28 \text{ lb/KW-HR}$$

$E_b = 0.678$

$f_b = 1.0$

$f_p = 0.995$

$f_t = 1.015$

$$SR = \frac{TSR}{(E_b)(f_b)(f_p)(f_t)} = \frac{7.28}{(.678)(1.0)(.995)(1.015)} = 10.65$$

Correction for load factor: Each generator (1500 KW) loaded down @ 1000 KW connected

$$\frac{1000}{1500} = 66.8\% \quad \therefore \; f_L = .952$$

$$\text{Final Steam Rate} = \frac{SR}{S_L} = \frac{10.65}{.952} = 11.2 \text{ lb/KW-HR}$$

11.2 LB/KW-HR × 2,000 KW = 22,400 LB/HR required. Leak-off steam: 2 × 50 LB/HR = 100 LB/HR

Turbo Generator Steam = 22,500 LB/HR

NET STEAM AVAILABLE:

102,000 LB/HR

-22,500 LB/HR

79,500 LB/HR

c.2) SUPERCHARGE REQUIREMENTS--(Constant Line Pressure @ 450 psig)

Using MST-13 Turbine, 12 stage, steam conditions as above.

HP required: 7,750. SR for this turbine = 6.95 LB/BHP-HR (.76 efficiency)

$\therefore$ 6.95 LB/BHP-HR × 7,750 = 54,000 LB/HR

Leak off = 200 LB/HR + (.003 × BHP) = 223 LB/HR

Supercharger Steam = $\underline{\underline{54,223}}$ LB/HR required

NET STEAM AVAILABLE:

$$\begin{array}{r l} 79{,}500 & \text{LB/HR} \\ -\underline{54{,}223} & \text{LB/HR} \\ 25{,}277 & \text{LB/HR} \end{array}$$

c.3) MAIN FEED PUMP REQUIREMENTS--DISH @ 700 psig (Constant Line Pressure @ 450 psig)

Total flow to be pumped:

$$\begin{array}{r l} 102{,}000 & \text{LB/HR Supht} \\ \underline{15{,}000} & \text{LB/HR SAT} \\ 117{,}000 & \text{LB/HR TOTAL} \end{array}$$

$$117{,}000 \text{ LB/HR} \times \frac{1 \text{ HR}}{60 \text{ min}} \times \frac{1.198 \times 10^{-1} \text{ GAL}}{\text{LB}} = 234 \text{ GAL/Min}$$

η_p = EFF of pump = .66

$$\text{BHP} = \frac{144 \times \Delta P \times \nu_f \times W}{33{,}000 \times 60 \times \text{EFF.}} = \frac{\Delta P \text{ (GPM)}}{1715 \text{ (EFF)}} = \frac{700(234)}{1715\ (.66)} = 144.5 \text{ HP}$$

IHP ≡ Internal Turbine Horsepower

$$\text{IPH} = \frac{\text{BHP}}{\eta_m} = \frac{144.5}{.98} = 147.5, \eta_m = \text{Mechanical Efficiency}$$

Flow Required:

$$\dot{m} = \frac{\text{IHP}(550)(3600)}{\eta_{ei}(\Delta h)(778)} = \frac{147.5(550)(3600)}{(.5)(468.3)(778)}$$

where η_{ei} = Internal Turbine Eff.

Δh = 468.3

$$\begin{array}{r l} = 1{,}610 & \text{LB/HR} \\ \underline{50} & \text{Losses} \\ \underline{\underline{1{,}660}} & \text{Feed Pump Steam} \end{array}$$

c.4) NET STEAM FOR DUMPING TO MAIN CONDENSER:

25,277 LB/HR

-1,660 LB/HR

23,617 LB/HR

A.2 17-Knot Endurance Calculations (Figure 3.24)

Assumptions are the same as in A.1 except:

3.	Supercharger pressure ratio:	1.17:1
6.	G.T. Firing Temperature:	1230°F
8.	G.T. Pressure Ratio (80% Compressor speed):	5.084:1
9.	Base G.T. Flow Rate:	205 LBM/SEC
11.	G. T. Compressor Efficiency(Isentropic):	.72

a.) SUPERCHARGER WORK

@ $T_{00} = 530^\circ$ R., $P_{r00} = 1.640$ $h_{00} = 16.8$ Btu/lbm

$P_{00} = 14.7$ psia, $P_0 = 17.2$ lbf/in^2

$\therefore P_{r_0} = 1.17\,(1.640) = 1.920$

@ P_{r0}, $T_{0_s} = 555^\circ$ R., $h_{0_s} = 22.555$ Btu/lbm

$$\eta_{sc} = .85 = \frac{h_{0_s} - h_{00}}{h_{0_a} - h_{00}}$$

$\therefore h_{0_a} = 23.6$ Btu/lbm

$T_{0_a} = 98.5^\circ$ F.

$\dot{m}_{sc} = 1.17(205) = 240$ lbm/sec

$$W_{sc} = \dot{m}_{sc}(h_{0_a} - h_{00}) = \frac{(240)(6.8)}{.7074} = \underline{\underline{2,310 \text{ H.P.}}}$$

b.) GAS TURBINE WORK

Pressure drop through intercooler and ducting assumed to be same as that for full power. Temperature reduced to 70° F. in intercooler.

$P_1 = P_0 - .6 = 16.6$ lbf/in^2

$T_1 = 70^\circ$ F. $= 530^\circ$ R.

@ 70° F., $P_1 = 16.6$ lbf/in^2, $h_1 = 16.8$ Btu/lbm, $P_{r_1} = 1.640$

$P_2 = 5.084(16.6) = 84.5$ lbf/in^2 and $P_{r_2} = 1.64(5.084) = 8.34$

@ P_{r_2}, $T_{2_s} = 381.5^\circ$ F, $h_{2_s} = 91.8$ Btu/lbm

$$\eta = .72 = \frac{h_{2_s} - h_1}{h_{2_a} - h_1} \quad \therefore \; h_{2_a} = 120.8 \text{ Btu/lbm}$$

b.1) COMPRESSOR WORK

$$HP_c = \dot{m}_{GT}\Delta h = \dot{m}_{GT}\,(h_{2_a} - h_1) = \frac{1.17(205)(104.0)}{.7074} = \underline{\underline{36{,}200 \text{ HP}}}$$

Now assume all compressor work done by H.P. Turbine

$\therefore \; \Delta h_{HP} = 104.0$ Btu/lbm

Pressure loss in combustor, $P_{3_a} = 84.5\,(.97) = 82.0$ lbf/in^2

b.2) HIGH PRESSURE TURBINE

@ $T_{3_a} = 1230^\circ$ F, $h_{3_a} = 310.920$, $P_{r_3} = 112.794$

$h_{4_a} = h_{3_a} - \Delta h_{HP} = 439.920 - 104.0 = 206.92$ Btu/lbm

$T_{4_a} = 838.5^\circ$ F.

$$\eta_{HP} = .87 = \frac{h_{3a} - h_{4a}}{h_{3a} - h_{4s}} \quad \therefore \; h_{4_s} = 191.379 \text{ Btu/lbm}$$

@ h_{4_s}, $P_{r4_s} = 34.2$, $T_{4_s} = 778.5^\circ$ F.

Pressure Ratio: $\dfrac{112.79}{34.2} = 3.297 \;\therefore\; P_4 = \dfrac{82.0}{3.297} = 24.9$ lbf/in^2

b.3) LOW PRESSURE TURBINE

Gases now expand to 10" back pressure, 15.1 lbf/in^2

@ $T_{4_a} = 838.5^\circ$ F., $h_{4_a} = 206.920$

Pressure Ratio: $\dfrac{24.9}{15.1} = 1.649 \quad P_{r5_s} = \dfrac{40.9}{1.648} = 24.8$

@ 24.8, $h_{5_s} = 1652$ Btu/lbm, $T_{5_s} = 676.5^\circ$ F.

$$\eta_{LP} = .84 = \frac{h4_a - h5_a}{h4_a - h5_s} \quad \therefore \; h5_a = 171.920 \text{ Btu/lbm}$$

$T_{5_a} = 703.5^\circ$ F. (exhaust temperature)

b.4) WORK OUT TO SHAFT

Assume: a) Loss in power correction from 70° F. to 59° F. is 0.96 (Figure C.2)

b) Fuel/Air Ratio = 1.013

c) Gear Efficiency = 0.985

d) Loss in Flow from Ambient = .98

$$W_{LP} = \dot{m}_{LP}\Delta h = \frac{(240)(35)(.96)(1.013)(.985)(.98)}{.7074} = \underline{\underline{11,100 \text{ HP}}}$$

c.) STEAM CYCLE See Table 3.6 and Ref. (66)

Steam Conditions: P = 450 psig, T = 680° F., 39,650 LB/HR

c.1) TURBOGENERATOR REQUIREMENTS (51)--1140 KW/Ship (Constant line pressure @ 450 psig); or 570 KW/Engine Room (One generator)

Theoretical Steam Rate:

$$TSR = \frac{3412.1}{h_0 - h_p}$$

$h_0 = 1348.8$

$h_p = \underline{905.0}$ @ 1 psia

443.8

$$TSR = \frac{3412.1}{443.8} = 7.28 \text{ LB/KW-HR}$$

$E_b = 0.678$

$f_b = 1.0$

$f_p = 0.995$

$f_t = 1.005$

$$SR = \frac{TSR}{(E_b)(f_b)(f_p)(f_t)} = \frac{7.28}{(.678)(1.0)(.995)(1.005)} = 11.35$$

f_L: $\frac{570}{1500}$ = 37.9% Load

$\therefore\ f_t = .862$ Final S-R $= \frac{SR}{.862} = \frac{11.35}{.862} = 13.19$ lb/KW-hr

13.19 LB/KW-HR × 570 KW = 7,500 LB/HR

Leak-Off 1 × 50 LB/HR 50

Turbogenerator Steam 7,550 LB/HR required

NET STEAM AVAILABLE:

39,650 LB/HR

-7,550 LB/HR

32,100 LB/HR

c.2) SUPERCHARGER REQUIREMENTS--(Constant line pressure @ 450 psig) Using MST-13 Turbine, 12 stage, Impulse.

Steam conditions as above: HP required: 2,310 HP.

SR for this turbine = 6.95 LB/SHP-HR (.76 efficiency)

∴ 6.95 LB/SHP-HR × 2,310 = 16,050 LB/HR

Leak-Off 200 + (.003 × 2310) = 207

Supercharger Steam = 16,257 LB/HR Required

NET STEAM AVAILABLE:

32,100 LB/HR

-16,257 LB/HR

15,843 LB/HR

c.3) MAIN FEED PUMP REQUIREMENTS--DISH @ 700 psig (Constant line pressure @ 450 psig)

Total flow to be pumped: 39,650 Supht

15,000 LB/HR SAT

54,650 LB/HR TOTAL

$$54{,}650 \text{ LB/HR} \times \frac{1 \text{ HR}}{60 \text{ min}} \times 1.198 \times 10^{-1} \frac{\text{GAL}}{\text{LB}} = 109 \text{ GAL/MIN}$$

$$\text{BHP} = \frac{144 \times \Delta P \times \nu_f \times W}{33{,}000 \times 60 \times \text{EFF}} = \frac{\Delta P\ (\text{GPM})}{1717\ (\text{EFF})} = \frac{700(109)}{1715(.66)} = 68.5 \text{ HP}$$

$$\text{IHP} = \frac{\text{BHP}}{\eta_m} = \frac{68.5}{.98} = 70 \text{ HP}$$

Flow Required: $\Delta h = 443.8$

$$\dot{m} = \frac{\text{IHP}(550)(3600)}{\eta_{ei}(\Delta h)(778)} = \frac{70(550)(3600)}{(.5)(443.8)(778)} = 809 \text{ LB/HR}$$

+ 50 LB/HR Loss 50 LB/HR

Feed Pump Steam = 859 LB/HR Required

c.4) NET STEAM FOR DUMPING TO MAIN CONDENSER:

15,843

859

14,984 LB/HR

A.3 12-KNOT REPLENISHMENT CALCULATIONS (See Figure 3.25)

Assumptions are the same as in A.1 except:

3.	Supercharger pressure ratio:	1.13:1
6.	G.T. Firing Temperature	1110°F.
8.	G. T. Pressure ratio (80% compressor speed)	5.084:1
9.	Base G.T. Flow Rate:	205 LBM/SEC
11.	G.T. Compressor Efficiency (Isentropic):	.72

a.) SUPERCHARGER WORK

@ $T_{00} = 530°$ R., $P_{r_{00}} = 1.640$, $h_{00} = 16.8$ Btu/lbm

$P_{00} = 14.7$ psia, $P_0 = 16.60$ lbf/in^2

$\therefore P_{r_0} = 1.850$

@ P_{r_0}, $T_{0_s} = 548.5°$ R., $h_{0_s} = 21.210$ Btu/lbm

$$\eta_{sc} = .85 = \frac{h_{0_s} - h_{00}}{h_{0_a} - h_{00}}$$

$\therefore h_{0_a} = 21.0$ Btu/lbm

$T_{0_a} = 87.0°$ F.

$$\dot{m}_{sc} = 1.13(205) = 232 \text{ lbm/sec}$$

$$W_{sc} = \dot{m}_{sc}(h_{0_a} - h_{00}) = \frac{(232)(4.2)}{.7074} = \underline{\underline{1{,}380 \text{ HP}}}$$

b.) GAS TURBINE WORK

Pressure drop through intercooler and ducting assumed to be same as that for full power. Temperature reduced to 70° F. in intercooler.

$P_1 = P_0 - .6 = 16.0 \text{ lbf/in}^2$

$T_1 = 70° \text{ F.} = 530° \text{ R.}$

@ 70° F., $P_1 = 16.0 \text{ lbf/in}^2$, $h_1 = 16.8$ Btu/lbm, $P_{r_1} = 1.640$

$P_2 = 5.084\ (16.0) = 81.3 \text{ lbf/in}^2$ and $P_{r_2} = 5.084\ (1.64) = 8.34$

@ P_{r_2}, $T_{2_s} = 381.5°$ F., $h_{2_s} = 91.8$ Btu/lbm

$$\eta = .72 = \frac{h_{2_s} - h_1}{h_{2_a} - h_1} \qquad \therefore\ h_{2_a} = 120.8 \text{ Btu/lbm}$$

b.1) COMPRESSOR WORK

$$HP_c = \dot{m}_{GT}\Delta h = \dot{m}_{GT}(h_{2_a} - h_1) = \frac{1.13(205)(104.0)}{.7074} = \underline{\underline{34{,}100 \text{ HP}}}$$

Now assume all compressor work done by H.P. turbine

$\therefore\ \Delta h_{HP} = 104$ Btu/lbm

Pressure loss in combustor, $P_{3_a} = 81.3\ (.97) = 78.7 \text{ lbf/in}^2$

b.2) HIGH PRESSURE TURBINE

@ $T_{3_a} = 1110°$ F., $h_{3_a} = 278.090$, $P_{r_3} = 84.525$

$h_{4_a} = h_{3_a} - \Delta h_{HP} = 278.090 - 104.0 = 174.090$ Btu/lbm

$T_{4_a} = 710.5°$ F.

$$\eta_{HP} = .87 = \frac{h_{3_a} - h_{4_a}}{h_{3_a} - h_{4_s}} \qquad \therefore\ h_{4_s} = 158.5 \text{ Btu/lbm}$$

@ h_{4_s} $Pr_{4_s} = 22.736$, $T_{4_s} = 649°$ F.

Pressure ratio: $\frac{84.525}{22.736} = 3.72 \therefore P_4 = \frac{78.7}{3.72} = 21.2$ lbf/in^2

b.3) LOW PRESSURE TURBINE

Gases now expand to 10" back pressure, 15.1 lbf/in^2

@ $T_{4_a} = 710.5°$ F., $h_{4_a} = 174.090$

Pressure ratio: $\frac{21.2}{15.1} = 1.405$ $\quad Pr_{5_s} = \frac{27.705}{1.405} = 19.75$

@ 19.75, $h_{5_s} = 148$ Btu/lbm, $T_{5_s} = 608°$ F.

$\eta_{LP} = .84 = \frac{h_{4_a} - h_{5_a}}{h_{4_a} - h_{5_s}} \quad \therefore h_{5_a} = 152.09$ Btu/lbm

$T_{5_a} = 624.5°$ F. (exhaust temperature)

b.4) WORK OUT TO SHAFT

Assume: a) Loss in power correction from 70° F. to 59° F. is .96 (Figure C.2)

b) Fuel/air ratio = 1.01

c) Gear efficiency = 0.985

d) Loss in flow from ambient = .98

$$W_{LP} = \dot{m}_{LP}\Delta h = \frac{(232)(22)(.96)(1.01)(.985)(.98)}{.7074} = \underline{\underline{6,750 \text{ HP}}}$$

c.) STEAM CYCLE--See Table 3.6 and Reference (66)

12-KNOT REPLENISHMENT HEAT BALANCE (HIGH KW LOAD ALONGSIDE)

Steam Conditions: P = 450 psig, T = 620° F., 24,930 lb/hr

Assume: a) Heating steam secured

b) 1 × 1000 KW Diesel on the line

c.1) TURBOGENERATOR REQUIREMENTS--3230 KW/SHIP (less 1000 diesel) = 2230 KW/SHIP (Constant Line pressure @ 450 psig) or 1115 KW/Engine Room (one generator)

Theoretical Steam Rate:

$h_0 = 1314.6$

$h_p = \underline{885.0}$ @ 1 psia

429.6

$$TSR = \frac{3412.1}{h_0 - h_p} = \frac{3412.1}{429.6} = 7.95$$

$E_b = 0.678$

$f_b = 1.0$

$f_p = 0.995$

$f_t = 0.991$

$$SR = \frac{TSR}{(E_b)(f_b)(f_p)(f_t)} = \frac{7.95}{(.678)(1.0)(.995)(.991)} = 11.89$$

$$\text{Final S.R.} = \frac{SR}{f_L} \quad \frac{11.89}{.964} = 12.30 \text{ lb/KW-Hr}$$

f_L: $\frac{1115}{.500} = 74.3\%$

$\therefore f_L = 0.964$

12.30 lb/KW-HR 1115KW = 13,700 lb/hr

Leak off steam = $\underline{\quad 50}$ lb/hr

Turbogenerator steam = $\underline{\underline{13,750}}$ required

NET STEAM AVAILABLE:

24,930

$\underline{13,750}$

11,180

c.2) SUPERCHARGE REQUIREMENTS--(Constant line pressure @ 450 psig) Using MST-13 Turbine, 12 stage.

Steam conditions as above. HP required: 1,380 HP

SR for this turbine = 6.95 lb/BHP-HR (.76 efficiency)

$\therefore$ 6.95 LB/BHP-HR × 1,380 = 9,600 LB/HR

Leak off = 200 + (.003 × 1380) = 204

= $\underline{\underline{\text{9,804 LB/HR}}}$ required

NET STEAM AVAILABLE:

11,180 LB/HR

9,804 LB/HR

1,376 LB/HR

c.3) MAIN FEED PUMP REQUIREMENTS--DISCHG @ 700 psig (Constant line pressure @ 450 psig)

Total flow to be pumped:

24,930 LB/HR Supht

10,500 LB/HR SAT

35,430 LB/HR

$$35{,}430 \text{ LB/HR} \times \frac{1\text{HR}}{60 \text{ min}} \times 1.198 \quad 10^{-1} \frac{\text{GAL}}{\text{LB}} = 70.8 \text{ GPM}$$

$$\eta_p = .66$$

$$\text{BHP} = \frac{144 \times \Delta P \times \nu_f \times W}{33{,}000 \times 60 \times \text{EFF}} = \frac{\Delta P(\text{GPM})}{1715(\text{EFF})} = \frac{700(70.8)}{1715\ (.66)} = 43.8 \text{ HP}$$

$$\text{IHP} = \frac{\text{BHP}}{\eta_m} = \frac{43.8}{.98} = 44.8$$

Flow Required:

$$\dot{m} = \frac{\text{IHP}(550)(3600)}{\eta_{ei}(\Delta h)(778)} = \frac{44.8(550)(3600)}{(.5)(429.6)(778)} = 530 \text{ LB/HR}$$

+ 50 lb. Loss 50

Feed pump Steam = 580 LB/HR

c.4) NET STEAM FOR DUMPING TO MAIN CONDENSER:

1,376 LB/HR

- 580 LB/HR

796 LB/HR

A.4 Determination of Supercharger Compressor Characteristics

a.) BASIC APPROACH. As stated in Section 3.2.2, the two-stage axial flow characteristics were generated by combining two M.I.T. gas turbine laboratory curves (Nos. 6 & 7) for a single-stage axial compressor.

Looking at Figure 3.4 (P_2/p, vs $\dot{m}/\dot{m}^*$), the reasoning was to use two stages operating at a pressure ratio of 1.185 (i.e., 1.185 × 1.185 = 1.4, the desired pressure ratio) and get a value of $\dot{m}/\dot{m}^*$. This value of $\dot{m}/\dot{m}^*$ was used to go to Figure 3.5 (T_2/T_1 vs $\dot{m}/\dot{m}^*$), and obtain a value of T_2/T_1. A mass flow parameter was developed which represented the exit conditions from the first stage. That is, at exit a value called m_2 was developed:

$$m_2 = \frac{\dot{m}/\dot{m}^*\sqrt{T_2/T_1}}{P_2/P_1}$$

This gave the mass flow parameter for the inlet conditions to the second stage and enabled one to relate the two stages in the development of the supercharger compressor characteristics. Points on the original P_2/P_1 vs $\dot{m}/\dot{m}^*$ could now be picked off the curve at different speed parameters and replotted on a new curve. Points were chosen as follows: 1) near the surge line; 2) at midpoint, and 3) near the choke line along curves of non-dimensional speed parameter, π.π is defined as:

$$\pi = \frac{\text{tip speed}}{\text{velocity of sound at inlet}}$$

b.) SAMPLE CALCULATION

From P_2/P_1, vs $\dot{m}/\dot{m}^*$ at a pressure ratio of 1.185, $\dot{m}/\dot{m}^* = 0.234$ and $P_2/P_1 = 1.0555$ (Figure 3.5) at exit

$$m_2 = \frac{\dot{m}/\dot{m}^*\sqrt{T_2/T_1}}{P_2/P_1} = \frac{.236\sqrt{1.0555}}{1.185} = 0.203$$

Now considering a point near surge, $\dot{m}/\dot{m}^* = 0.21$, $P_2/P_1 = 1.214$ and $T_2/T_1 = 1.068$.

$m_2{}^1$ defined as second-stage inlet mass flow parameter

$$m_2{}^1 = \frac{0.21\sqrt{1.214}}{1.068} = 0.179$$

Multiplying this value by $\dot{m}/\dot{m}^*$ and dividing by m_2 gave outlet mass flow parameter, m_3.

That is
$$m_3 = \frac{m_2 1}{m_2} \times \dot{m}/\dot{m}^* = \frac{0.179}{0.203} \times .234 = 0.206$$

at this point $P_2/P_1 = 1.216$ and $T_2/T_1 = 1.067$. The pressure ratio thus is $1.214 \times 1.216 = 1.475$, which is greater than 1.4 as expected.

Thus a point on the new compressor characteristic was $\dot{m}/\dot{m}^* = 0.210$ @ $P_2/P_1 = 1.475$. The constant speed lines, π, were interpreted such that the .8 speed line became 1.0 speed (maximum) for the new characteristics, and the remaining speed lines were scaled accordingly. After all points were plotted (see Table A.1) the $\dot{m}/\dot{m}^*$ was converted to real flow in pounds per hour. The final supercharger characteristics are shown in Figure 3.3.

Efficiency was determined as follows:

at design point:

$$\frac{T_{out}}{T_{in}} = T_2/T_1 = 1.0555 \times 1.0555 = 1.115136$$

$$\therefore\ T_{out} = 1.115136 \times 530^\circ\ \text{R.} = 591.02208^\circ\ \text{R.} \qquad 591.02208$$

$$\underline{-460} \qquad \text{and} \qquad \underline{-530}$$

$$131^\circ\ \text{F.} \qquad\qquad 61.02208^\circ\ \text{R.}$$

$$\eta = \frac{(\Delta T/T_{in})\text{isen}}{(\Delta T/T_{in})\text{ act}} = \frac{(55/530)}{(61.02208/530)} = .901313$$

TABLE A.1 SUPERCHARGER CHARACTERISTIC DETERMINATION

Base Design Parameters @ 100% RPM:

$\dot{m}/\dot{m}^* = 0.234$, $P_2/P_1 = 1.185$, $T_2/T_1 = 1.056$

$M_2 = 0.203$, $\dot{m} = 348.5$ LB/HR

% RPM	(1) $\dot{m}/\dot{m}^*$	(2) P_2/P_1	(3) T_2/T_1	(4) $\sqrt{T_2/T_1}$	(5) m_2	(6) $m_2{}^1$	(7) m_3	(8) New P_2/P_1	(9) (2)× (8)	(10) $\dot{m}$, LB/HR
100% - Surge	.192	1.219	1.075	1.037	.203	.1635	.1885	1.219	1.480	285.9
100% - Mid	.210	1.214	1.067	1.033	.203	.1785	.2060	1.216	1.475	362.8
100% - Choke	.247	1.158	1.051	1.025	.203	.2186	.2520	1.148	1.330	367.9
87.5% - Surge	.166	1.166	1.057	1.028	.203	.1464	.1688	1.166	1.359	247.2
87.5% - Mid	.190	1.154	1.047	1.023	.203	.1680	.1940	1.151	1.328	282.9
87.5% - Choke	.216	1.122	1.038	1.019	.203	.1960	.2260	1.130	1.268	321.7
75% - Surge	.144	1.126	1.043	1.021	.203	.1310	.1510	1.123	1.264	214.5
75% - Mid	.170	1.112	1.035	1.018	.203	.1560	.1798	1.104	1.228	253.1
75% - Choke	.189	1.094	1.029	1.015	.203	.1750	.2020	1.078	1.179	281.4
62.5% - Surge	.122	1.090	1.031	1.015	.203	.1136	.1309	1.087	1.185	181.7
62.5% - Mid	.140	1.082	1.025	1.013	.203	.1310	.1510	1.073	1.161	208.5
62.5% - Choke	.160	1.064	1.020	1.011	.203	.1520	.1750	1.044	1.111	238.2
53.8% - Surge	.098	1.062	1.023	1.012	.203	.0930	.1072	1.062	1.128	145.9
53.8% - Mid	.120	1.057	1.018	1.009	.203	.1145	.1319	1.048	1.108	178.7
53.8% - Choke	.139	1.042	1.014	1.007	.203	.1343	.1548	1.022	1.065	207.0

d.) SUPERCHARGER SIZING

Assumed: 1) $r_h/r_t = 0.55$

2) $c_x = 500$ ft/sec (inlet velocity)

3) $u_t = 1000$ ft/sec max (blade tip speed)

4) $\dot{m} = 348.5$ lb/sec

$$\dot{m} = \rho A C_x \qquad \rho = \frac{P}{RT} = \frac{17.6\ \mathrm{lbflbm^\circ R(144)in^2}}{\mathrm{in^2 53.34\ ft\ lb_f\ 563^\circ R}}$$

$$\rho = \frac{17.6\ \mathrm{lb_f\ lbm\ ^\circ\ R}\ (14.4 \times 10^1\ \mathrm{in}^2)}{\mathrm{in}^2\ (53.34\ \mathrm{ft\ lb_f})(5.63 \times 10^2\ ^\circ\ \mathrm{R})\mathrm{ft}^2}$$

$$= .859 \times 10^{-1} = .0859\ \mathrm{lb_m/ft^3}$$

$$\therefore\ A = \frac{\dot{m}}{\rho c_x} = \frac{348.5\ \mathrm{lb_m\ ft^3\ sec}}{\mathrm{sec}\ 8.59 \times 10^{-2}\ \mathrm{lb_m}\ 5.0 \times 10^2\ \mathrm{ft}} = 8.13\ \mathrm{ft}^2$$

$$A = \pi(r_t{}^2 - r_h{}^2) \qquad r_h = .55r_t$$

$$= \pi[r_t{}^2 - (.55r_t{}^2)]$$

$$= \pi[r_t{}^2\ (1 - .55^2)]$$

$$= \pi[r_t{}^2\ (1-.303)]$$

$$= \pi(r_t{}^2\ .697)$$

$$8.13\ \mathrm{ft}^2 = \pi(.697\ r_t{}^2)$$

$$r_t{}^2 = \frac{8.13}{3.14(6.97 \times 10^{-1}} = .371 \times 10^1 = 3.71\ \mathrm{ft}^2$$

$$r_t = 1.93\ \mathrm{ft}$$

$$u_t = \frac{2\pi N r_t}{60} \qquad \therefore\ N = \frac{u_t 60}{2\pi r_t}$$

$$N = \frac{1000\ \mathrm{ft}\ (6.0 \times 10^1)}{\mathrm{sec}\ (6.28)\ (1.93)\mathrm{ft}} = \underline{\underline{4950\ \mathrm{RPM}}}$$

A.5 Condenser Design (Reference 67)

a.) GIVEN: Steam to be condensed: 117,000 LB/HR

Tube size = 5/8" 18 gauge

Material = 90-10 CuNi

Exhaust = 2" Hg abs.

Single pass (n = 1) scoop injection

$\Delta h = 921 - 69.7 = 851.3$ Btu/lb

T_1 = Injection temperature = 67° F.

L = Effective Tube Length: = 10.32 feet

V = 9 ft/sec (water velocity)

F_1 - Cleanliness correction factor = 0.85

F_2 = Material correction factor for U = .90

F_3 = Inlet water correction factor for U due to cooling water less than 70° F. = .99

b. SOLUTION

b.1) U, BASIC HEAT TRANSFER COEFFICIENT IN BTU/HR-FT^2 ° F.

$$U \equiv 270\sqrt{V}$$

$$\therefore U = 270\sqrt{9} = 810 \text{ Btu/Hr-ft}^2 \text{ ° F.}$$

U_c, corrected heat transfer coefficient:

$$U_c = UF_1F_2F_3 = 810(.85)(.90)(.99) = \underline{\underline{615 \text{ Btu/Hr-Ft}^2 \text{ ° F.}}}$$

b.2) TEMPERATURE RELATIONSHIPS

$$a \equiv \frac{U_cLk}{500\ V} = \frac{615(10.32)(.2407)}{500\ (9)} = 0.340$$

Where: L = Effective length of each tube in feet per pass no. of passes

k = tube constant, s/g

$$e^a = 1.404$$

s = square feet of external tube surface per foot of length

g = G.P.M. per tube at one foot per second water velocity

$$T_0 \equiv T_s - \frac{T_s - T_1}{e^a}$$

$$= 101.14 - \frac{101.14 - 67}{1.404}$$

$$= 101.14 - 24.3 = \underline{\underline{76.84° \text{ F.}}}$$

Where: T_0 = outlet (overboard) circulating water temp., °F.

T_s = saturated steam temp., at condenser pressure

b.3) CONDENSING SURFACE AND NO. OF TUBES

Where: G = Gallons of circulating water/min.

W = Steam condensed, lbs/hr

500 = Conversion factor for flow rate and circulating water density

$$G \equiv \frac{W\Delta h}{500(T_0 - T_1)}$$

$$= \frac{117{,}000(851.3)}{500(9.84)}$$

$$= 14{,}500 \text{ GPM}$$

Where: A = Total condensing surface based on O.D. tube area, ft^2

$$\text{and } A \equiv \frac{LGk}{V} = \frac{(10.32)(14{,}500)(.2407)}{9}$$

$$= \underline{\underline{4000 \text{ ft}^2}}$$

Where: N = Total number of tubes per water pass

$$N = \frac{G}{gV} \quad \frac{14{,}500}{(.68)(9)} = \underline{\underline{2{,}370 \text{ Tubes}}}$$

b.4) CONDENSER SIZE

Assumed: Area of tube holes in tube sheet does not exceed 24% of the total tube sheet area before drilling.

$$\text{Area per tube} = \frac{\pi d^2}{4} = \frac{3.14(5/8)^2}{4} = .7854(5/8)^2 = 0.3068 \text{ in}^2$$

Area of Holes - 2,370 × .3068 = 723 in^2

Minimum Tube Sheet Area: $\frac{723}{.24} = 3020 \text{ in}^2 = 20.9 \text{ ft}^2$

For water box depth not less than one-half tube sheet diameter but not greater than 45 in.

For circular tube sheet

$$d^2 = \frac{20.9}{.7854} = 26.6 \text{ ft}^2$$

$$d = 5.15 \text{ ft}$$

Water box depth $= \frac{5.15}{2} = 2.57$ ft

Total Length of Condenser:

$$L + 2\,(\text{water box depth}) = 10.32 \text{ ft} + 2(2.57 \text{ ft})$$

$$= 10.32 + 5.15$$

$$= 15.47 \text{ ft} \sim 15'\ 6''$$

Total Width of Condenser: 6.48 ft ∿ 6' 5"

Total Height of Condenser: 6.48 ft ∿ 6' 5"

b.5) HOTWELL VOLUMETRIC CAPACITY

Steam Rate/min $= \frac{117{,}000}{60} = 1{,}950$ lbs/min

Specific volume ν of saturated condensate @ 101.14

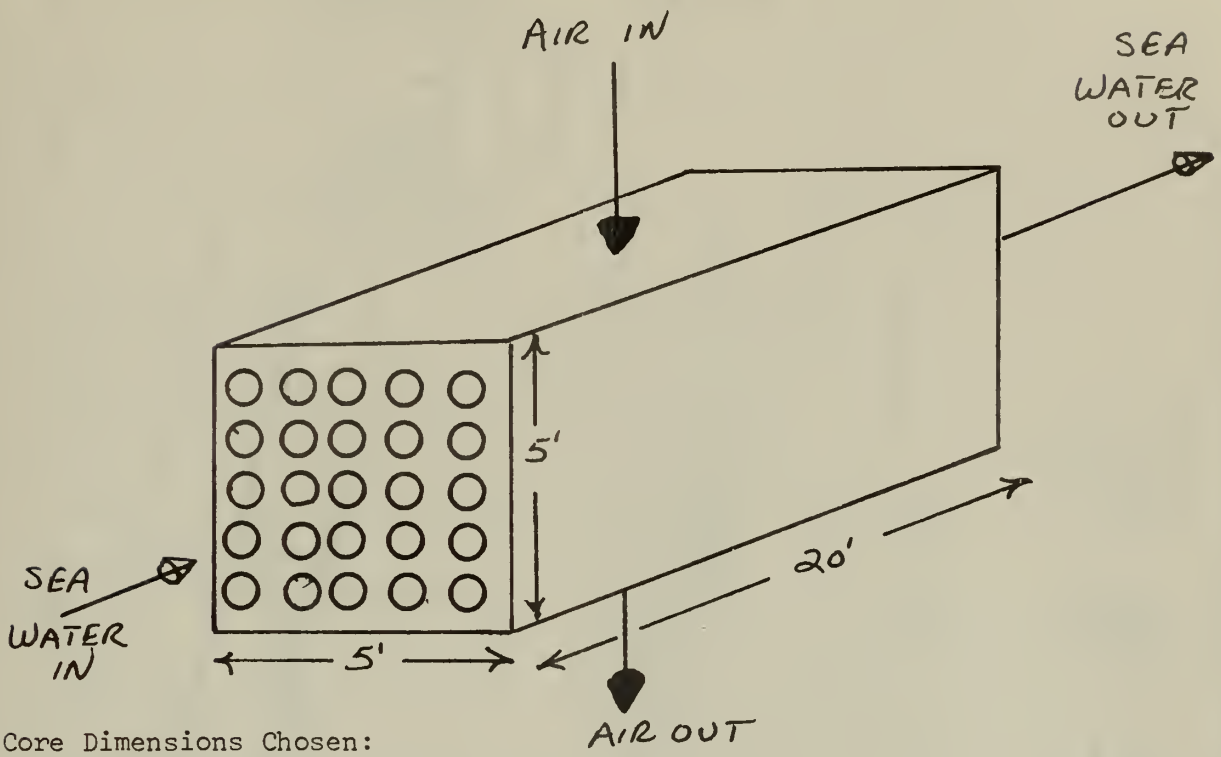
$$= .0161 \text{ ft}^3/\text{lb}$$

∴ Hotwell volumetric capacity.

$$= 1{,}950 \text{ lb/min} \times 1.61 \quad 10^{-2} \frac{\text{ft}^3}{\text{LB}}$$

$$= \underline{\underline{31.40 \text{ ft}^3/\text{min}}}$$

APPENDIX B--INTERCOOLER DESIGN (FULL POWER CONDITION) (27)



Core Dimensions Chosen:

a) Air side frontal area, $A_{fR} = 5 \times 20 = 100$ ft^2

b) Water side frontal area, $A_{fR_W} = 5 \times 5 = 25$ ft^2

c) Total volume = $25 \times 20 = 500$ ft^3

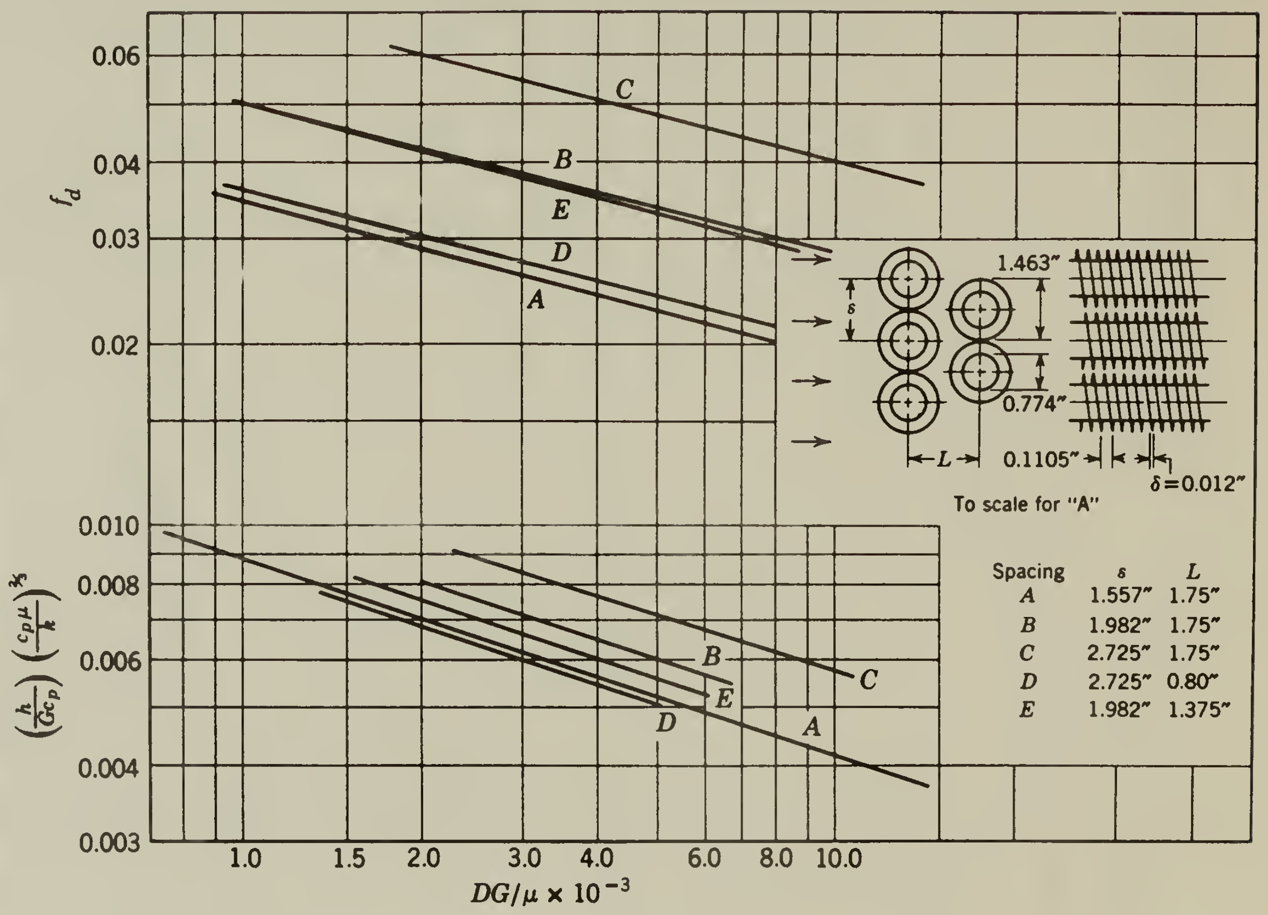
TYPE: Finned circular tube--Jameson Data

Continuous fin CF 9.05-3/4 J-E Kays & London (27)

See Figure B.1 (68)

AIR SIDE

Air flow rate	= 1,277,000 lb/hr
Air humidity	= .03 lb H_2O/lb air
Air entering, temp. °F.	= 131° F = 591° R
Air entering, press. psia	= 20.6 psia
Air leaving, temp. °F.	= 70° F = 530° R



SURFACES CF-9.05-3/4J

Tube OD	0.774 in.
Fin pitch	9.05/in.
Fin thickness	0.012 in.
Fin area/total area	0.835

	A	*B*	*C*	*D*	*E*
Air passage equivalent diameter	0.0168	0.0269	0.0445	0.0159	0.0211 ft
Free flow area/frontal area	0.455	0.572	0.688	0.537	0.572
Heat-transfer area/total volume	108	85.1	61.9	135	108 ft^2/ft^3

Figure B.1 Intercooler Design--Finned Tube Characteristics (68)

1. Surface Characteristics--Table 14 (27)

Flow passage hydraulic radius $4r_{ha} = 0.02108$ ft

$r_{ha} = 0.00527$ ft

Total air side transfer area/total volume $\alpha_a = 108$ ft^2/ft^3

Free flow area/frontal area $\sigma_a = 0.572$

Fin area/total area $= 0.835$

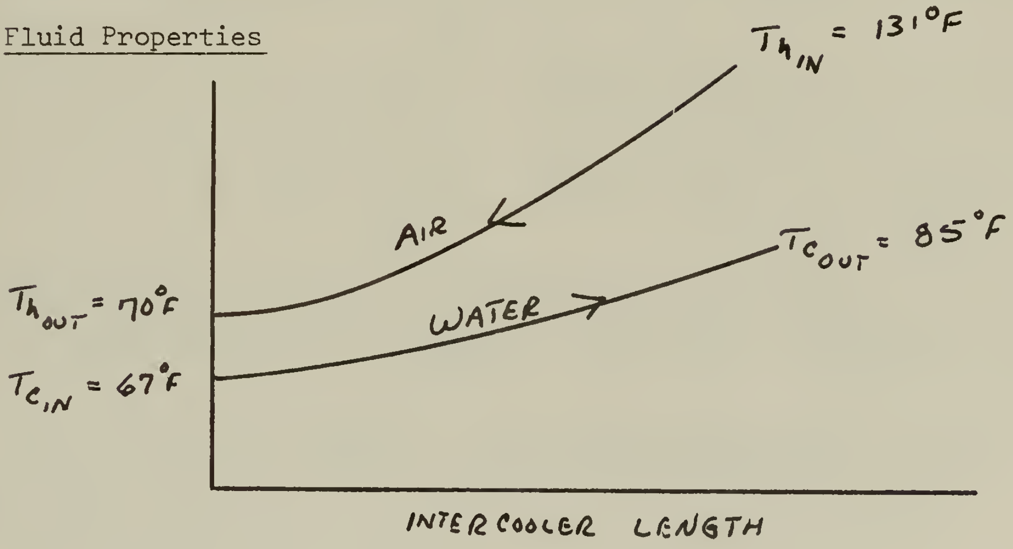
Fin metal thickness $\delta = 0.012\text{in}/12 = \delta = 0.001$ ft

Fin material, aluminum $k = 100$ Btu/hr-ft °F

Fin length, 1/2 distance between tubes, $\zeta = 0.2595$ in

$$\frac{(1.982 - 1.463)}{2} = .2595 \text{ in}$$

2. Fluid Properties



Average air temp., T_a

$$T_a = \frac{T_{c_{in}} + T_{c_{out}}}{2} + \frac{(T_{h_{in}} - T_{c_{out}}) - (T_{h_{out}} - T_{c_{in}})}{\ln \dfrac{(T_{h_{in}} - T_{c_{out}})}{(T_{h_{out}} - T_{c_{in}})}}$$

$$= \frac{67 + 85}{2} + \frac{46 - 3}{\ln 46/3} = 76 + \frac{43}{\ln 15.3} = 76 + \frac{43}{2.72}$$

$$= 76 + 15.8 = 91.8° \text{ F}$$

with this T_a average, Fig. 114 (27)

Viscosity, $\mu = 0.046$ lb/hr ft

Prantl No. $P_r = 0.700$, $P_r^{2/3} = .7883$

Specific ht, $C_p = 0.240$ Btu/lb ° F

Correct specific heat for humidity: Fig. 116 (27)

$X_{cw} = 1.025 \times .240 = .246$ Btu/lb ° F

Now assume 2% pressure drop, outlet air pressure is:

$20.600 - .02(20.600) = 20.188$ psia

Correct specific volumes at inlet and outlet for humidity, Fig. 116 (27)

1 = Inlet Where: $X_{aw} = 0.98$

2 = Outlet $\nu_1 = \frac{1}{P_1} = \frac{1\ RT_1}{X_{aw}\ P_1} = \frac{53.345 \times 591}{0.98 \times 20.6 \times 144} = 10.844 \text{ ft}^3/\text{lb}$

$$\nu_2 = \frac{1}{P_2} = \frac{1\ \ RT_2}{X_{aw}\ P_2} = \frac{53.345 \times 530}{0.98 \times 20.188 \times 144} = 9.924 \text{ ft}^3/\text{lb}$$

3. Reynolds No., Stanton No., and Friction Factor and h_a

$$G = \dot{m}/\text{free flow area} = \frac{\dot{m}}{A_{fr} \times \sigma_a} = \frac{1,277,000}{100 \times .572} = \frac{1,277,000}{57.2} = 22,390 \text{ lb/hr ft}^2$$

$$\text{Reynolds No., } R_e = \frac{4r_{h_a}\ G}{\mu} = \frac{(21.08 \times 10^{-3})(22,390)}{4.6 \times 10^{-2}} = 10,220$$

From Figure 98 (27)

Friction factor = f = 0.028

and $\frac{h}{GC_p}(P_r^{2/3}) = .0046$ $S_t = \frac{h}{GC_p}$, Stanton No.

$$\therefore S_t = \frac{46 \times 10^{-4}}{7.88 \times 10^{-1}} = 5.84 \times 10^{-3} = .00584$$

$$h_a = S_t \times G \times C_p = (5.84 \times 10^{-3})(22.39 \times 10^{3})(2.46 \times 10^{-1}) = 32.2$$

Btu/hr ft^2 °F.

WATER SIDE

Water flow rate $\dot{m}_w$ = 3,000,000 lb/hr˜6,000 GPM

Water entering, Temp., ° F. = 67° F.

Water leaving, Temp., ° F. = 85° F.

1. Surface Characteristics

Outside dimensions per tube = 0.774" ID

Frontal area associated with one tube (1.982 × 1.375) = 2.725 in^2

Free flow area of one tube, $\pi\frac{d}{4}$ = 0.60759 in.

Inside periphery of one tube, πD, = 2.43 in.

Free flow/frontal area $\frac{.60759}{2.725\ \text{in.}^2}$ σ_w = 0.2229

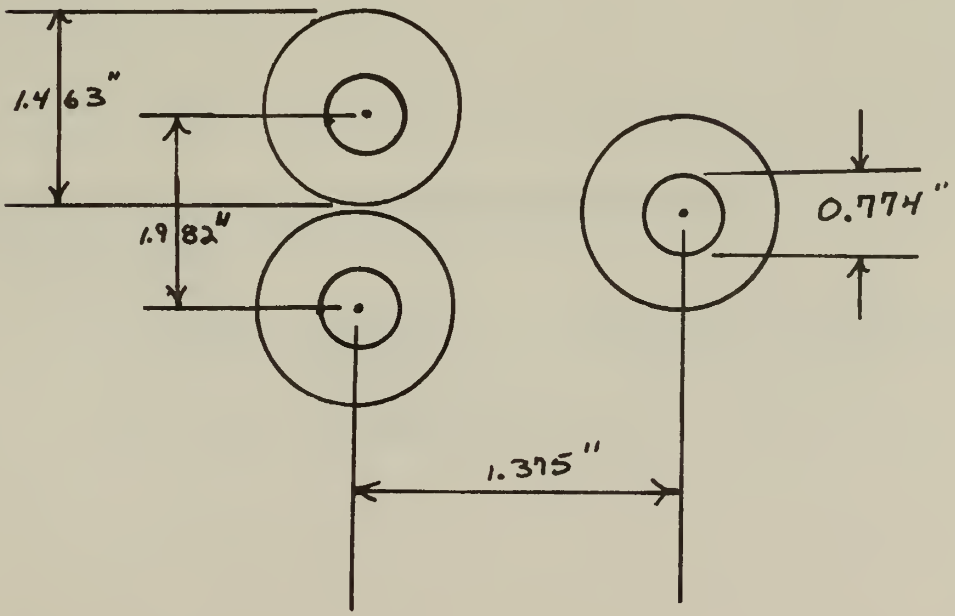
No. of tubes = 3600 in^2/frontal area = 3600/2.725 = 1321

Water side transfer area/total volume

$$= \frac{\text{periphery} \times \text{length} \times \text{no. of tubes}}{\text{vol}} = \frac{2.43}{12} \times \frac{20 \times 1321}{500} = 10.7 = \alpha_w$$

Water side flow passage hydraulic radius, r_{h_w} = D/4 = .1935 in.

= .01615 ft.



2. Fluid Properties

Average water temp., $T_w = \frac{67 + 85}{2} = 76$

with this temp. ((27), Fig. 115)

Viscosity, $\mu = 2.3$ lb/hr ft.

Prantl No. $P_r = 6.3$

Specific HT, $C_p = 1.0$ Btu/lb°F.

Specific vol., $\nu = 1/64.0 = .01565 \text{ ft}^3/\text{lb}$

3. Reynolds No., Stanton No., Friction Factor and h

$$G = \frac{\dot{m}_w}{\text{free flow area}} = \frac{\dot{m}_w}{A_{fr_w} \times \sigma_w} = \frac{3{,}000{,}000}{25 \times .2229} = 5.40 \times 10^5 \text{ lb/hr ft}^2$$

$$R_e = \frac{4rh_w \times G}{\mu} = \frac{4(1.615 \times 10^{-2})(5.40 \times 10^5)}{2.3} = 15.2 \times 10^3 = 15{,}200$$

From Figure 28 (27)

$$S_t \left(\frac{\mu_w}{\mu_m}\right)^{.20} \times 10^4 = 14$$

μ_w = viscosity of water

μ_m = mean viscosity

as $\frac{\mu_w}{\mu_m} \rightarrow 1.0$, $S_t = .0014$

From Figure 29 (27)

Friction factor, $f = 0.0068$

and $h_w = S_t \times G \times C_p = (14 \times 10^{-4})(5.40 \times 10^5)(1.0) = 755 \text{ Btu/hr-ft}^2\ {}^\circ\text{F}$.

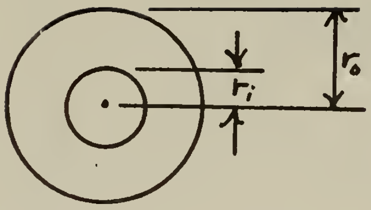
Fin Efficiency and Overall Surface Effectiveness

1. Fin Efficiency

$$m = \frac{\sqrt{2(h_w)}}{k\delta} = \frac{\sqrt{2(32.2)}}{1\times10^2(1\times10^{-3})} = \sqrt{64.4 \times 10^1} = \sqrt{644} = 25.4$$

$$m \times \zeta = 25.4 \times \frac{2.595 \times 10^{-1}}{12} = .548$$

$$r_o = \frac{1.463}{2} = .733 \qquad \frac{r_o}{r_i} = \frac{.733}{.386} = 1.89$$



$$r_i = \frac{.744}{2} = .386$$

From Figure 1 (27)

$$\eta_{fin} \cong 0.88$$

2. Overall Surface Efficiency

$$\eta_o = (1 - \frac{A_f}{A})(1 - \eta_f) = 1 - (.835)(1 - .88)$$

$$1 - (8.35 \times 10^{-1})(12 \times 10^{-2})$$

$$1 - 100 \times 10^{-3} = 1 - .1$$

$$\eta_o = 0.900$$

Heat Transfer Coefficient and Intercooler Effectiveness

$$\frac{1}{U_a} = \frac{1}{\eta_o h_a} + \frac{1}{\frac{\alpha_w}{\alpha_w h_w}} = \frac{1}{.90(32.2)} + \frac{1}{\frac{10.7}{108}(755)} = .0343 + .01335 = .04765$$

$$U_a = \frac{100 \times 10^{-2}}{4.765 \times 10^{-2}} = 21.0 \text{ Btu/hr-ft}^2 \, {}^\circ\text{F.}$$

C_a = 1,277,000 × .246 = 315,000 (air side capacity rate) = C_{min}

C_w = 3,000,000 × 1.0 = 3,000,000 (water side capacity rate) = C_{max}

$$A_a = \alpha_a \times \text{Vol}$$

$$\text{NTU} = \frac{U_a A_a}{C_{min}} = \frac{108(500)(21.0)}{3.15 \times 10} = 3.6 \text{ and } \frac{C_{min}}{C_{max}} = \frac{3.15 \times 10^5}{3.0 \times 10^6} = .105$$

From Figure B.2 (27),

Effectiveness, ε, = 0.95

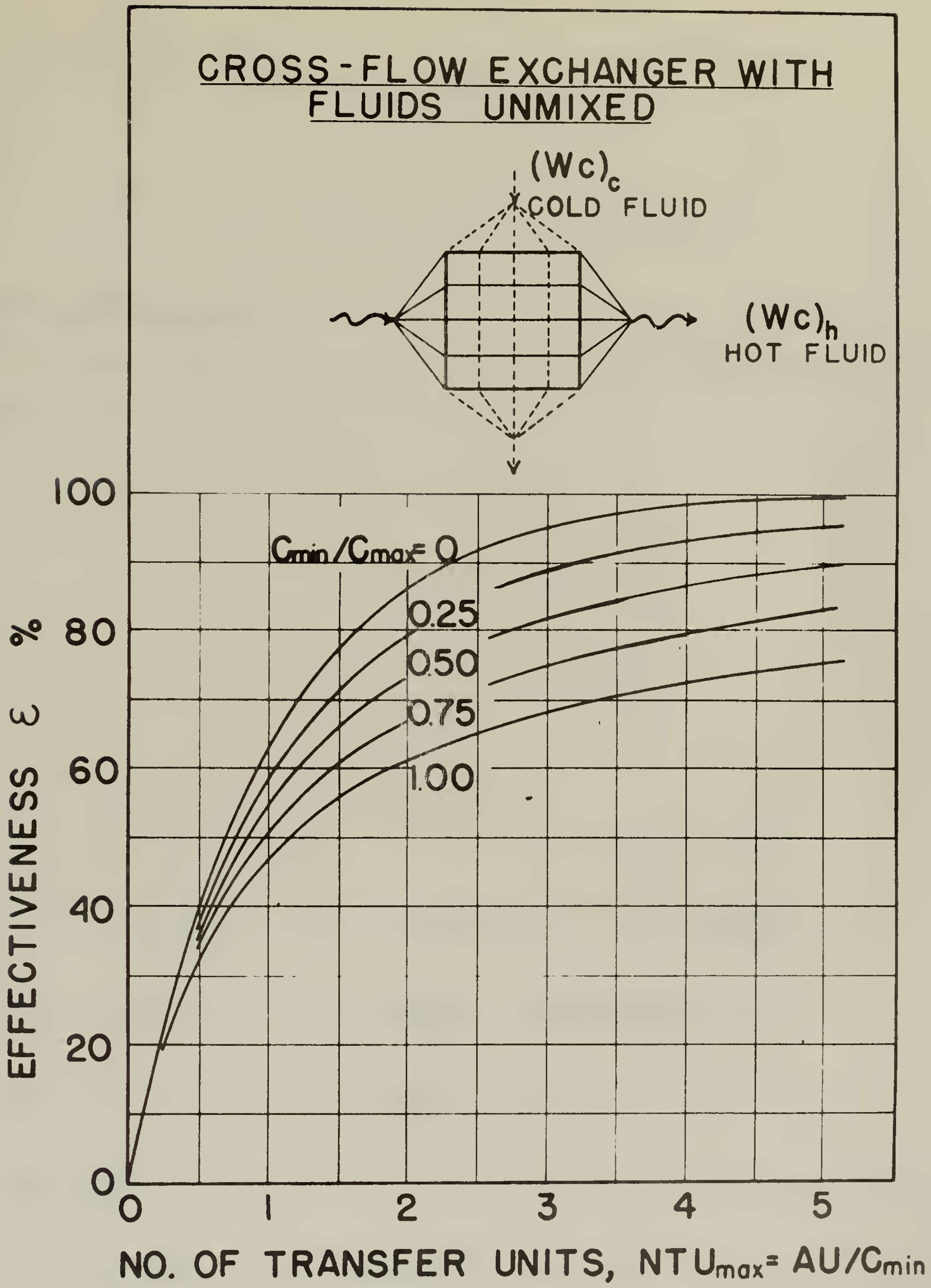


Figure B.2 Intercooler Design--No. of Transfer Units (27)

(NTU versus Effectiveness)

$$E = \frac{C_a(T_{h_{in}} - T_{h_{out}})}{C_a(T_{h_{in}} - T_{c_{in}})} = \frac{131 - T_{h_{out}}}{131 - 67} = .95$$

$$T_{h_{out}} = 131 - .95(64) = 131 - 60.8$$

$T_{h_{out}} = 70.2°$ F. (Temperature into gas turbine)

Pressure Drop, Air Side

For flow normal to tube banks, entrance and exit losses are neglected and the pressure drop is:

$$\frac{\Delta P}{P_1} = \frac{G^2}{2g_c}\,\frac{\nu_1}{P_1}\left\{ \underset{\text{(flow accelerator)}}{(1 + \sigma_a)\left(\frac{\nu_2}{\nu_1} - 1\right)} + \underset{\text{(core friction)}}{f\,\frac{\Delta}{A_c}\left(\frac{\nu_m}{\nu_1}\right)} \right\}$$

$$\frac{A}{A_c} = \frac{L}{r_h} = \frac{20}{12 \times 5.27 \times 10^{-3}} = .317 \times 10^3 = 317$$

$$1 + \sigma^2 = 1 + (.572)^2 = 1.326$$

$$\nu_m = \frac{53.345 \times 580}{.98 \times 20.3 \times 144} = 10.65 \text{ ft}^3/\text{lb}$$

$$\frac{\Delta P}{P_1} = \left(\frac{22.39 \times 10^3}{3.6 \times 10^2}\right)\left(\frac{10.844}{6.44 \times 10^1}\right)\left(\frac{1}{(20.6)(144)}\right)\left\{[(1.326)\left(\frac{9.924}{10.844} - 1\right)]\right.$$

$$\left. + [(2.8 \times 10^{-2})(3.17 \times 10^2)\,\frac{10.65}{10.844}]\right\} = \frac{(38.6)(1.685 \times 10^{-1})}{(20.6)(144)}$$

$$\left\{[(1.326)(-9.5 \times 10^{-2})] + [8.7]\right\} = \left(\frac{65.1 \quad 10^{-1}}{(20.6)(144)}\right)\left\{-12.6 \quad 10^{-2} + 8.7\right\}$$

$$\frac{(65.1 \times 10^{-1}}{(20.6)(144)}(7.574) = 16.95 \times 10^{-3} = .01695 \text{ or } 1.695\% \text{ pressure drop}$$

NET GAS PRESSURE DROP THROUGH INTERCOOLER: $20.6 \text{ psia} \times 1.695 \times 10^{-2}$

$$= 34.9 \times 10^{-2} = \underline{\underline{.349 \text{ psia}}}$$

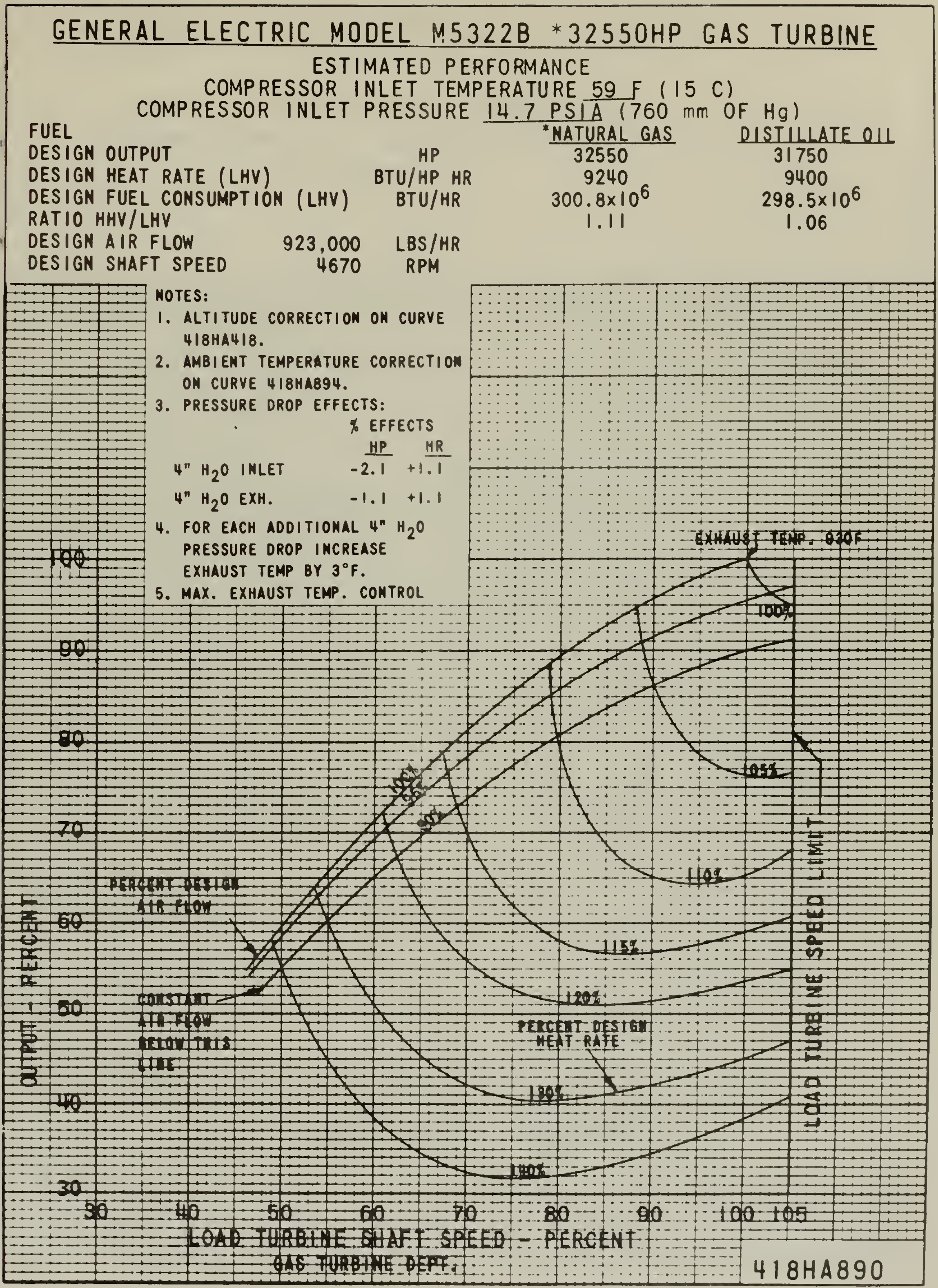
APPENDIX C. DETAILED DESCRIPTION OF THE GENERAL ELECTRIC MS 5322(B) GAS TURBINE

General Description (24)(60)(67)(70)

The General Electric gas turbine Model 5322 (B) has basic characteristics as shown in Figures C.1 and C.2 (60). It is an industrial heavy-duty two-shaft marine engine with variable angle load turbine nozzles. The gas turbine consists of a 16-stage axial flow compressor, combustion chamber and wrapper, and a single high pressure turbine coupled directly to the compressor shaft. The high pressure turbine discharges into a two-stage low pressure (load) turbine which is coupled directly to a controllable and reversible propeller via a locked train reduction gear arrangement. The load turbine has variable angle nozzles strator vanes each of which passes through the casing and is connected via linkage to the other nozzles--thus all can be rotated simultaneously. Rotating the vanes to increase nozzle area drops the inlet pressure and decreases the energy passed on to the load turbine, while closing the nozzle increases the energy to the power turbine. The great advantage of the variable angle nozzle for use aboard ships is the ability to keep firing temperature at a maximum by automatically adjusting the variable nozzle to reduce air flow. With a heat recovery steam generator placed after the gas turbine, it is desirable to keep exhaust temperature high to gain maximum heat recovery. This can promote lower fuel consumption at off design power levels. See Figure c.3 (60)

1. Compressor (24)

The compressor is a 16-stage axial flow design, with air flow from inlet to discharge. Air is compressed in small increments in the space between the rotor and stator blades, that is, alternate rotating



GENERAL ELECTRIC MODEL M5322B *32550HP GAS TURBINE

ESTIMATED PERFORMANCE
COMPRESSOR INLET TEMPERATURE 59 F (15 C)
COMPRESSOR INLET PRESSURE 14.7 PSIA (760 mm OF Hg)

FUEL			*NATURAL GAS	DISTILLATE OIL
DESIGN OUTPUT		HP	32550	31750
DESIGN HEAT RATE (LHV)		BTU/HP HR	9240	9400
DESIGN FUEL CONSUMPTION (LHV)		BTU/HR	300.8×10^6	298.5×10^6
RATIO HHV/LHV			1.11	1.06
DESIGN AIR FLOW	923,000	LBS/HR		
DESIGN SHAFT SPEED	4670	RPM		

NOTES:

1. ALTITUDE CORRECTION ON CURVE 418HA418.
2. AMBIENT TEMPERATURE CORRECTION ON CURVE 418HA894.
3. PRESSURE DROP EFFECTS:

	% EFFECTS HP	% EFFECTS HR
4" H_2O INLET	-2.1	+1.1
4" H_2O EXH.	-1.1	+1.1

4. FOR EACH ADDITIONAL 4" H_2O PRESSURE DROP INCREASE EXHAUST TEMP BY 3°F.
5. MAX. EXHAUST TEMP. CONTROL

Figure C.1 General Electric Model 5322 (B)--Estimated Performance (60)

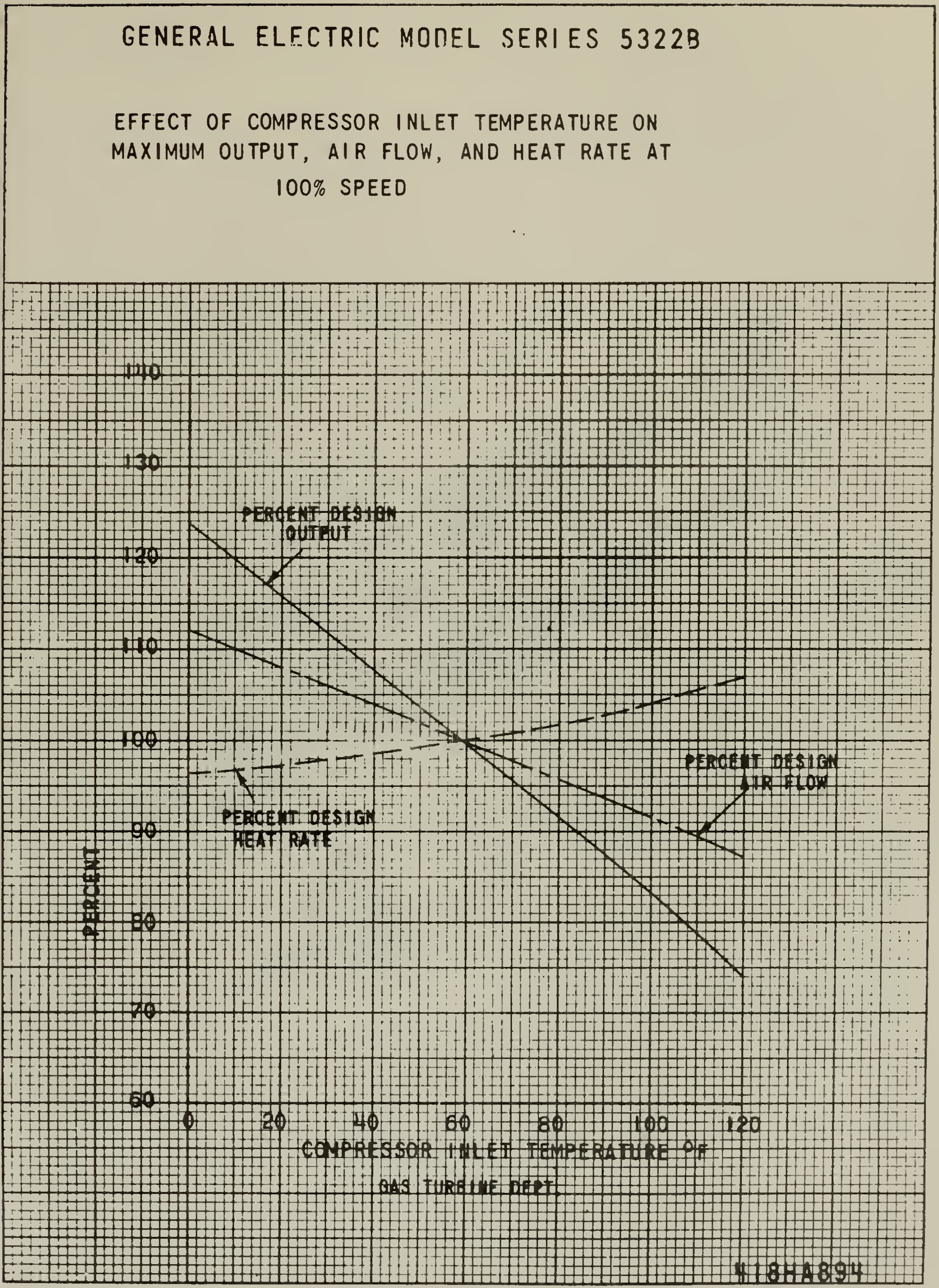
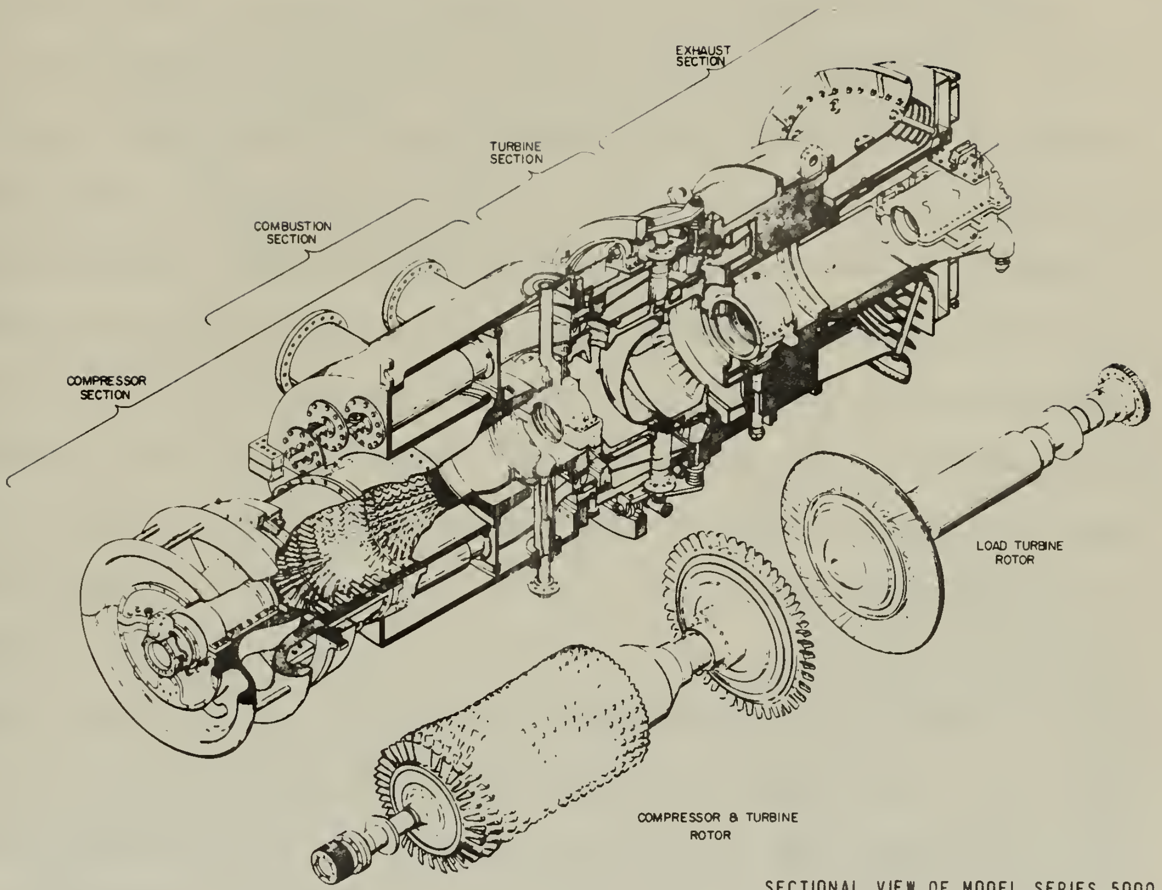


Figure C.2 General Electric Model 5322 (B)--Effect of Compressor Inlet Temperature (60)



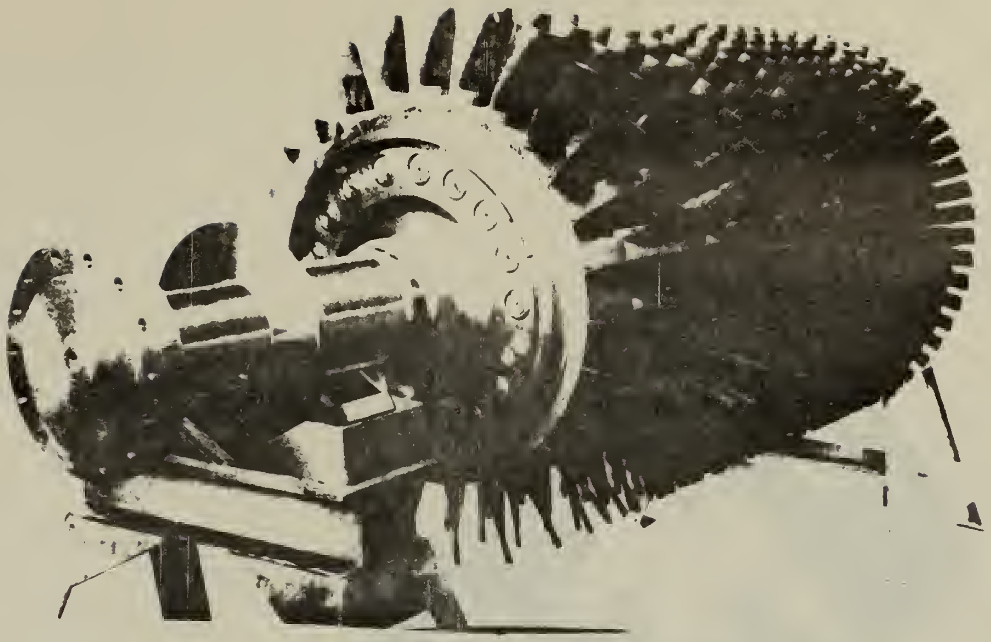
SECTIONAL VIEW OF MODEL SERIES 5000
TWO-SHAFT GAS TURBINE

Figure C.3 (60)

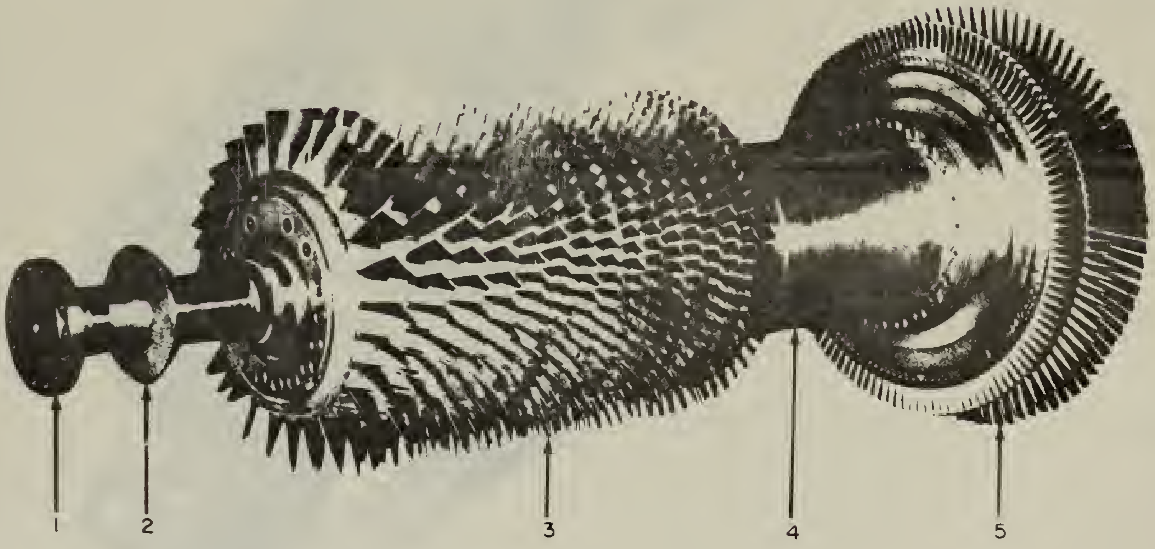
and stationary airfoil shaped blades. See Figures C.4 and C.5 (24)(71). The compressor from inlet to outlet has a pressure ratio of 8.2:1 at an RPM of 5100 and a mass flow of 348.5 (full power supercharged).

The inlet casing is designed to guide inlet air from the intercooler through a right-angled turn into the horizontal axial flow compressor. It is made of nodular cast iron and carbon steel, and contains the forward journal bearing (No. 1), high pressure Kingsbury thrust bearing, associated lube oil connections, as well as inlet guide vanes. The main part of the compressor casing is split horizontally and consists of a forward, after and outlet sections. The forward compressor casing contains the stator blading and rings for the first four compressor stages and bolts directly onto the inlet casing. It also has the extraction belt from the fifth stage used for air bleed during start-up and for sealing air when the compressor is up to speed. The after compressor casing contains the stator blades and ring assemblies for the middle six stages plus an eleventh stage extraction belt. The discharge casing of the compressor contains the final six stator blades and the exit guide vanes. Exit guide vanes help to convert the whirl velocity from the last compressor stage into a pressure rise. The air then passes to an inner barrel of the discharge casing which forms a circular diffuser and increases the pressure with an attendent loss of velocity. From here the air goes directly into the combustion wrapper. The discharge casing houses the after journal bearing (No. 2) and well as labyrinth seals to minimize leakage.

The stator structure is configured such that at the compressor discharge casing the casing itself branches into eight struts, located between combustion chambers. These struts connect to the forward end of



Compressor rotor assembly. (Typical.



1. Coupling flange
2. Thrust runner
3. Compressor rotor
4. Distance piece
5. Turbine rotor

Unit rotor assembly (compressor-turbine). (Typical)

Figure C.4 (24)

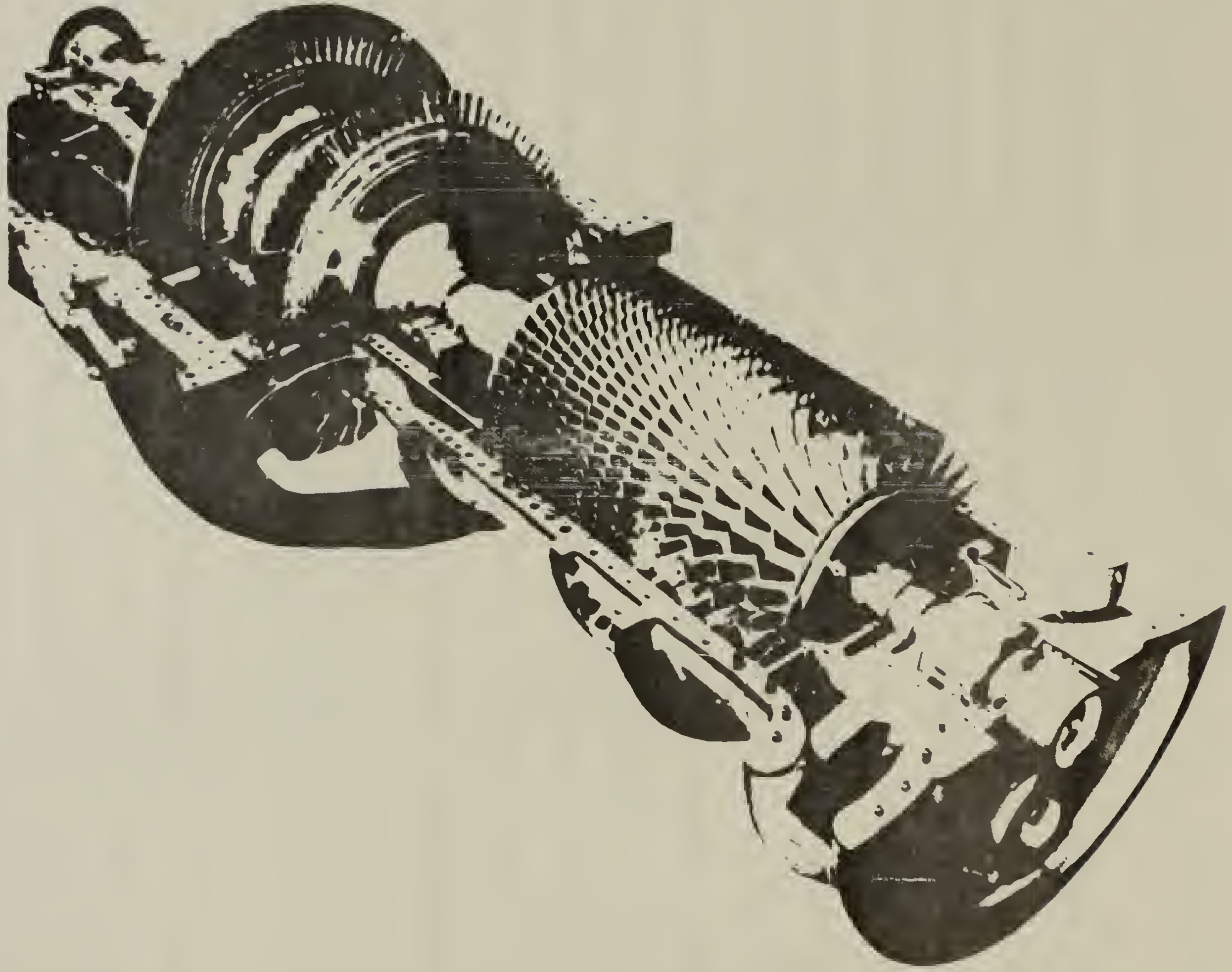


Figure C.5 High Pressure Section (71)

the turbine shell (casing) which extends downstream of the load turbine nozzle. Starting at the load turbine bucket inlet, the exhaust frame consists of a drum outside the exhaust diffuser, connected by six radial struts to an inner drum. The inner drum connects to the outboard load turbine bearing housing.

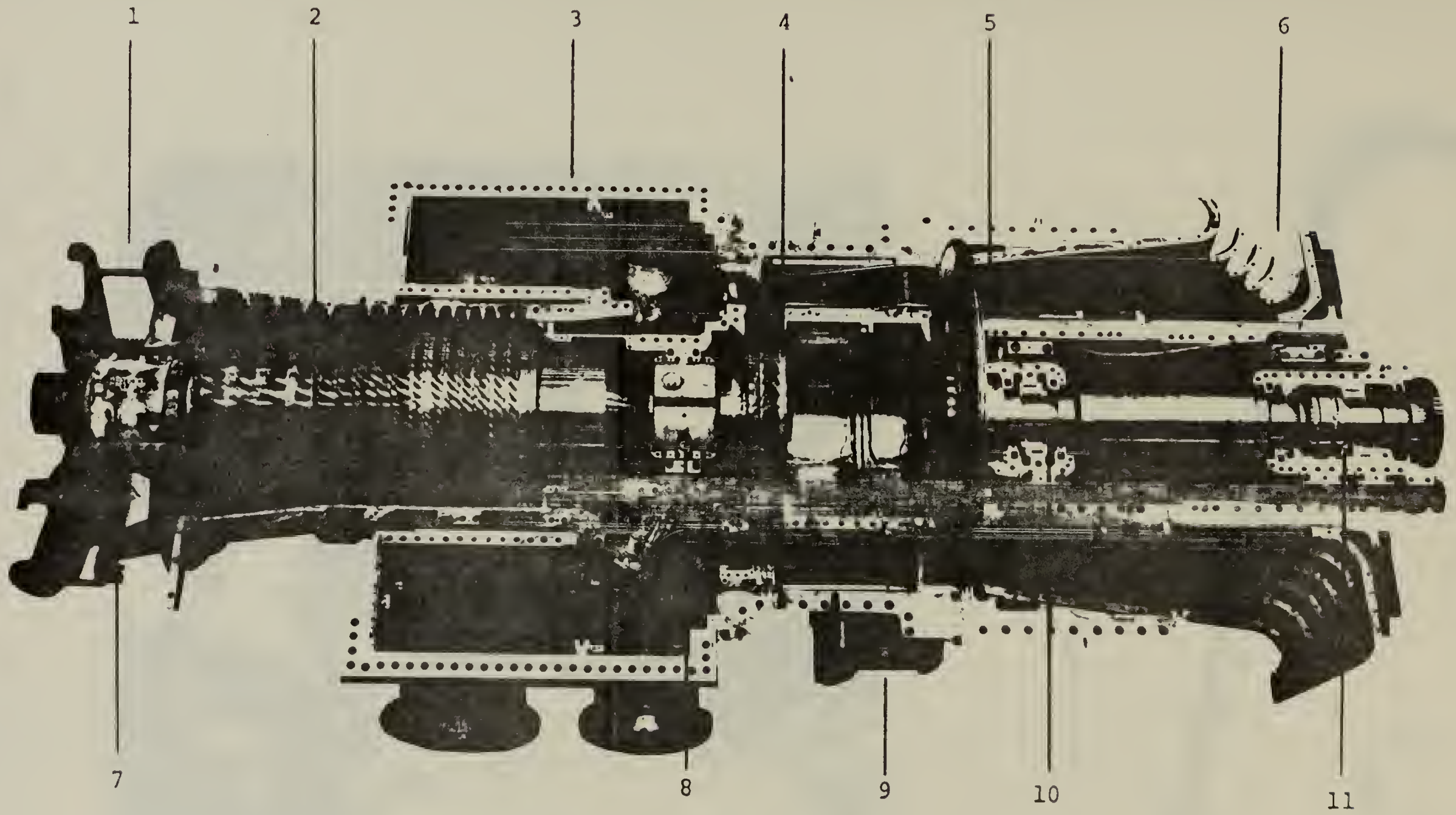
The compressor rotor consists of blade and wheel assemblies and two stub shaft assemblies. The wheels are rabbeted and held together by bolts arranged concentrically around the rotor axis. The bolt holes in the stub shafts are countersunk to reduce windage loss. The forward stub shaft contains the journal for the No. 1 bearing, the thrust runner for the high pressure thrust bearing, the sealing surfaces for the bearing oil deflectors, and the sealing surface for the compressor low-pressure air seal. The forward end of the stub shaft is machined for attachment of the accessory gear-to-compressor rotor flexible coupling. The aft stub shaft contains the journal for the No. 2 bearing and the sealing surfaces for the bearing oil deflectors. The forward end that bolts to the compressor rotor also forms the sealing surface for the compressor high pressure air seal, and the aft end is formed as a hollow cone, to which the first stage turbine wheel is bolted.

The airfoil shaped blades of both the compressor rotor and stator have dovetail shaped platforms by which they are retained in similarly shaped grooves in the stator blade rings and the compressor rotor wheels. The blade rings also have dovetail shaped bases and are retained in similarly shaped grooves in the casings. The blade ring segments are prevented from rotating in the grooves by long locking keys which fit into grooves machined in the horizontal joint flanges of the casings. The dovetails of

the rotor blades are peened in place.

2. Combustion System, Fuels and Fuel Treatment System (24)(70)

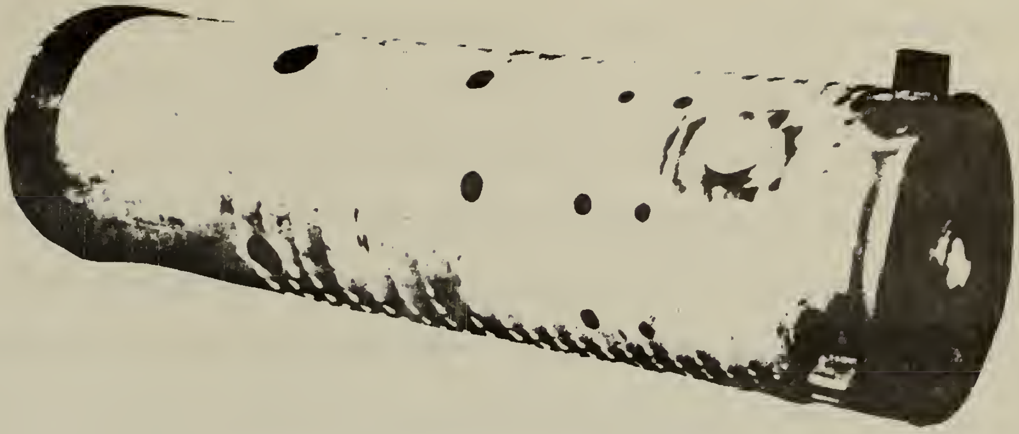
The combustion wrapper is a cylindrical fabrication split on the horizontal centerline, enclosing the area containing the combustion chambers and the transition pieces. The combustion chambers are in an annular space formed by the outer wrapper and an inner cylindrical wrapper surrounding the compressor casings. The annular space is closed by a wall containing openings for insertion of, and access to, the combustion chambers which are closed by cylindrical cover plates. Figure C.6 (24) shows the passage of air from the compressor to the combustion section. When the air reaches the annular space a portion of that air enters the inner cap and the combustion holes in the cap end of the lines--this is the primary combustion air. Another portion of the air enters the cooling louvers which are distributed over the surface of the cap and liner. This air serves to provide a film of relatively cool air along the inner surface of the cap and liner and protects these surfaces from the high temperature combustion gases by means of convection. The remainder of the air enters the the large mixing holes and mixes with the combustion gases. The combustion chambers consist of twelve 10-inch diameter chambers located around the outside of the compressor housing and enclosed in the combustion wrapper, as stated earlier. Fig. (C.7) (24) Fuel is introduced into the primary zone of the liner by a fuel nozzle which protrudes through the inner cap. The fuel nozzle gives support to the cap end of the liner and centers it in the outer casing. Combustion of the fuel and air mixture is initiated by spark plugs (two in number) located in separate chambers of the gas turbine unit. Crossfire tubes provide for near-immediate ignition



MM 5002 - Marine Heavy-Duty Gas Turbine
(Upper Casing Removed)

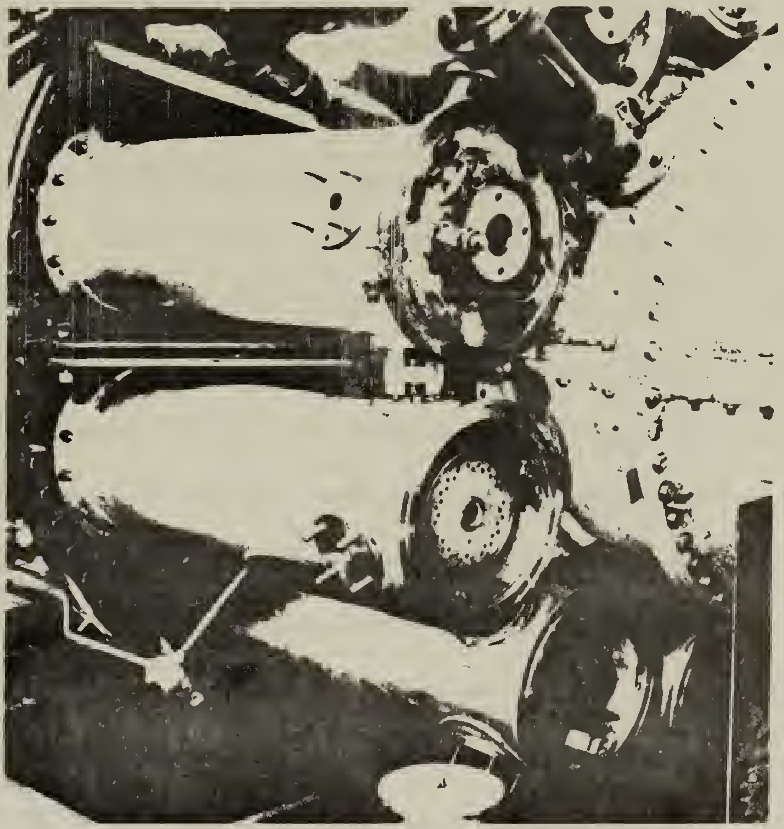
1. Inlet Casing
2. Compressor Rotor
3. Discharge Casing
4. High-Pressure Turbine
5. Low-Pressure Turbine
6. Exhaust Plenum
7. No. 1 Journal Bearing Assembly
8. No. 2 Journal Bearing Assembly
9. Turbine Shell and Second-stage Variable Nozzle Assmebly
10. No. 3 Journal and Thrust Bearing Assembly
11. No. 4 Journal and Thrust Bearing Assembly

Figure C.6 (24)



Typical combustion chamber liner.

(PP-06)



Combustion section showing combustors in various stages of assembly.

Figure C.7 (24)

to the remainder of the combustion chambers. There are also two ultraviolet flame detectors located in two separate combustion chambers which are different from the chambers which house the spark plugs. An indication of loss of flame by one of the detectors will sound an alarm, while a loss of flame indication to both detectors will shut the gas turbine down. There are also flame sight ports in the covers to visually check the combustion system. The hot gases pass through the chamber to the transition piece, which serves to conduct them to the annular turbine nozzle. The transition pieces are bolted firmly to the first stage nozzle ring by a single bracket so that the sides of the transition piece are free to move relative to the turbine nozzle ring in order to accommodate differential thermal expansion. Air seals are also used in the transition piece to prevent excess leakage.

As presently configured, this gas turbine will use Navy Distillate as the basic fuel as specified by the American Society for Testing and Materials. Navy Distillate has properties and contents not yet fully determined with regard to sodium and vanadium, and the 1.3 sulfur may present problems. The latter is important because of the waste heat boiler, and "cold end corrosion" may form as the sulphur combines with water vapor to form sulphuric acid. From this list of specifications, the most important to the high temperature, high performance are the vanadium and sodium contents. During the combustion process, vanadium's corrosive action is the destruction of the protective oxide layer that normally exists on the surface of heat resistant alloys, thereby exposing the base metal to attack. Sodium is dangerous insofar as it combines with sulfur to form sodium sulfate. When this acts with catalytic agents such as chlorine or carbon, sodium sulfate

attacks the metal grain boundaries. Often this attack takes place undetected until the turbine blades break catastrophically. In the event that these levels are too high, one may resort to the installation of the system shown in Figures C.8 and C.9 (72,73) which basically washes away sodium and calcium (also harmful) and brings the fuel oil down to acceptable levels. The vanadium may only be removed by chemical additives, usually magnesium sulphate. The addition of this fuel treatment system would give the ship the ability to burn residual fuels if distillate is not available. For purposes of analysis, plant layout, and costing, the system has not been included, and fuel will be treated only with centrifugal purifier.

3. High and Low Pressure (Load) Turbines (Fig C. 10) (24)

The high pressure turbine rated speed is 5100 RPM and the load turbine rated speed is 4,670, with capability of continuous operation at up to 4900 RPM. The turbine section of the gas turbine unit is made up of the following major component assemblies: the turbine shell, the exhaust hood frame and the exhaust hood skin, which encase the turbine rotors and serve as structural members of the unit assembly; the first and second stage nozzle assemblies which form the stationary portion of the gas path from the combustors to the exhaust hood and divide the energy of the combustion gases between the two turbine rotor assemblies; the second stage diaphragm assembly, which separates the first and second stages and provides the means of distributing cooling air to the inner faces of the turbine wheels; and finally, the two turbine rotor assemblies, consisting of high-pressure or first stage wheel and its buckets or blades which are bolted to the compressor aft stub shaft to form a high-pressure rotor assembly, and a low-pressure or second stage rotor. The first stage provides

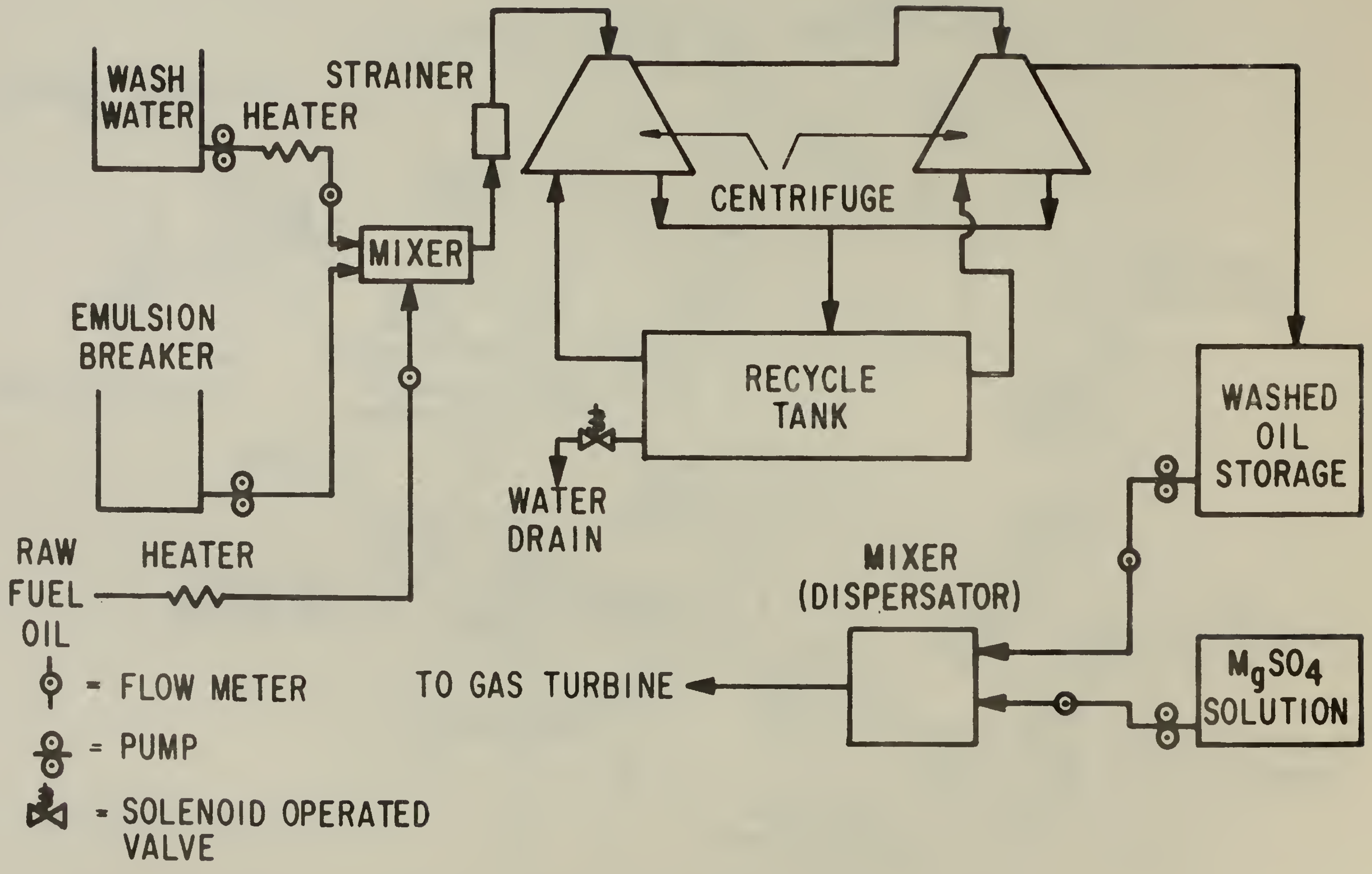
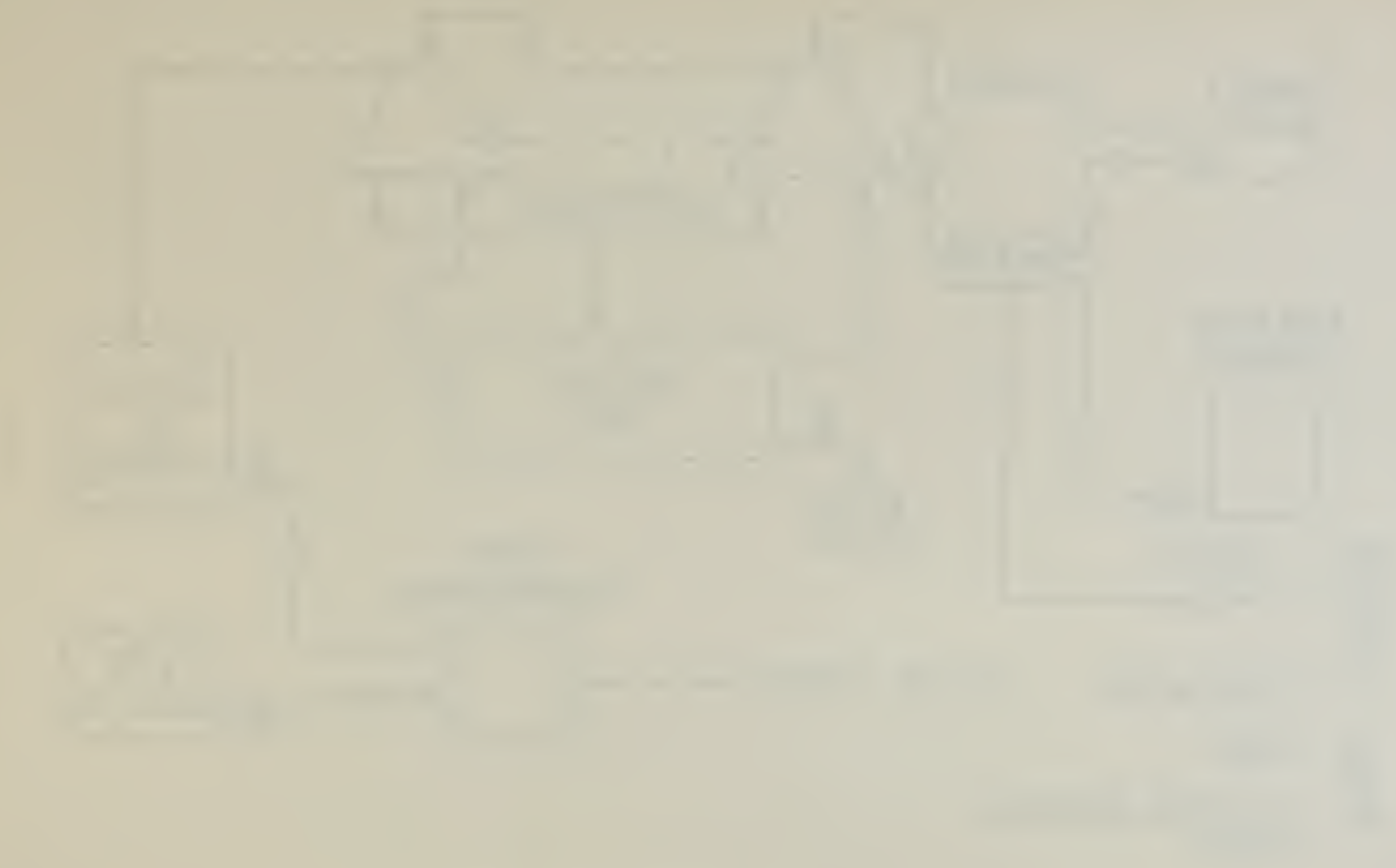


Figure C.8 Basic Fuel Treating System Schematic (72)



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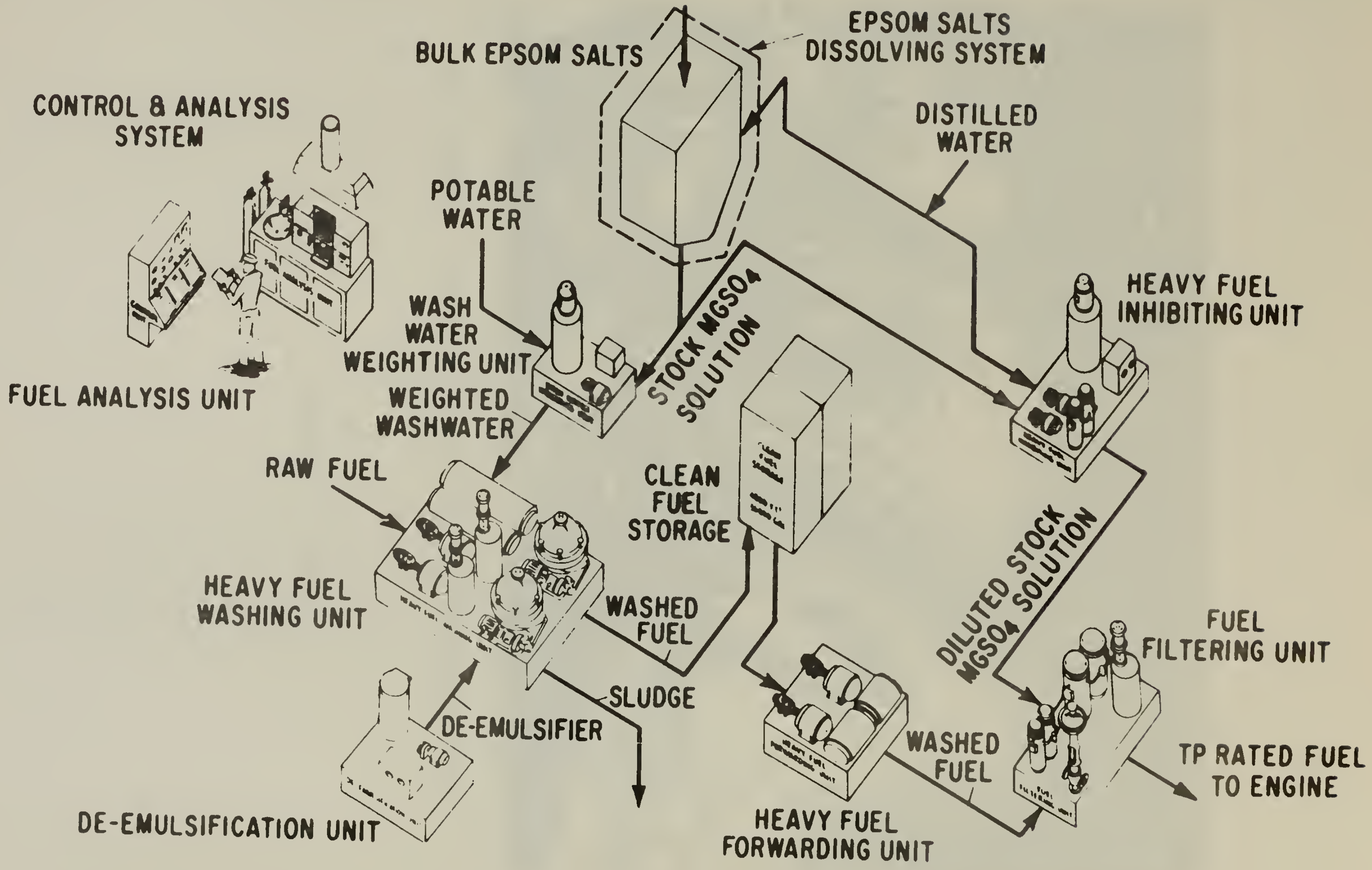
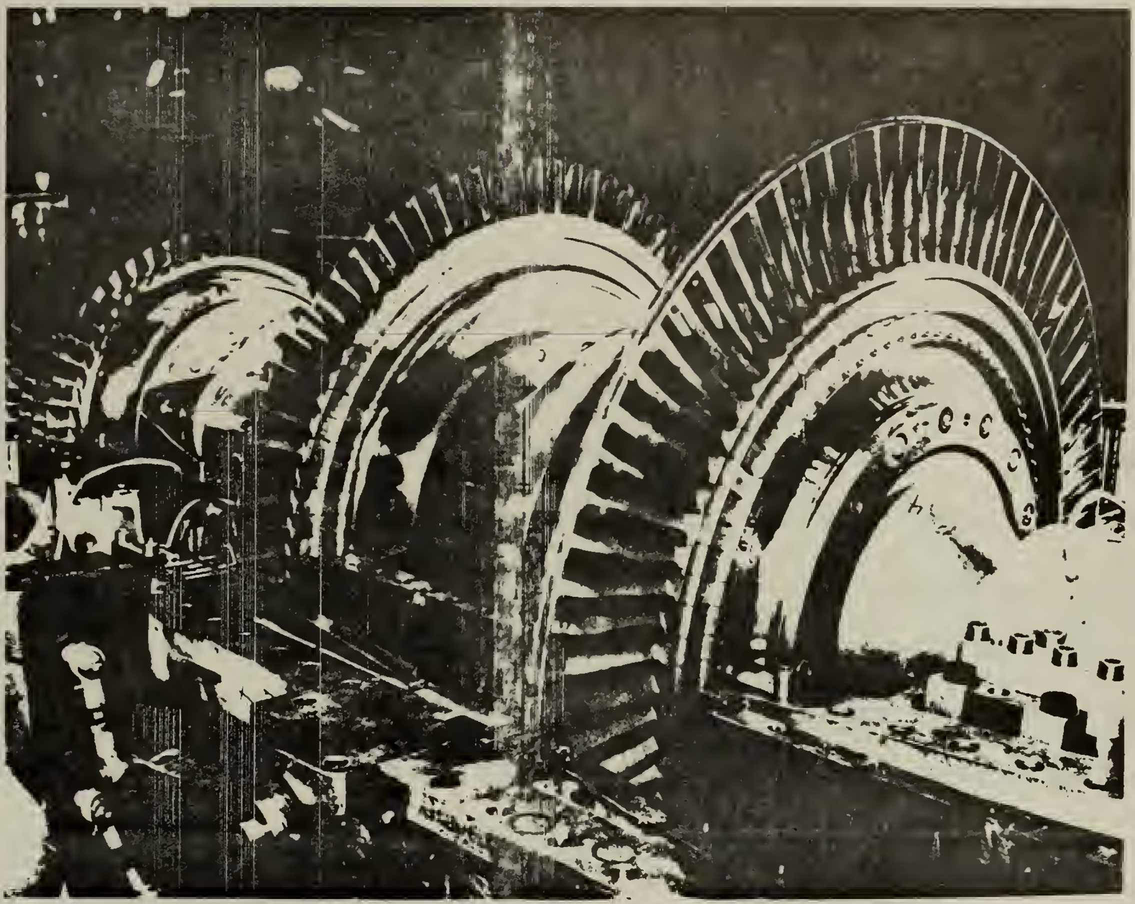


Figure C.9 Complete Residual Fuel Treating System (73)



GAS TURBINE -- HP ROTOR IS AT LEFT, LOAD TURBINE ON RIGHT
(TYPICAL)

Figure C.10 (24)

the power necessary for dividing the unit air compressor and the turbine driven accessories. The second stage rotor provides power out to the reduction gear, shafting, and ultimately to the propeller. It has been modified from the normal machine due to larger power required. The stator parts of the turbine section are split in half on the horizontal plane for ease of disassembly and reassembly for maintenance of the machine.

The first stage nozzle consists of twelve precision-cast segments. Each segment contains three vanes. Trailing edge cooling is obtained by compressor discharge air that passes in the vane core and then through holes onto the vane pressure surface immediately upstream of the trailing edge. All three stages utilize precision cast, long shank buckets. The long shank feature shields the wheel rim and bucket root fastening from the high temperature in the main gas stream, and adds vibration damping. The first stage bucket vane is hollowed out to reduce the centrifugal load and also to reduce transient thermal stresses. The second and third stage buckets use a tip shroud to provide vibration damping and to mount seal teeth that reduce the tip leakage flow.

The second stage nozzle consists of 32 precision cast nozzle vanes mounted on shafts that extend radially through the turbine casing. Each shaft carries a lever arm that connects to the control ring which rotates around the casing to vary the nozzle throat area. The control ring is actuated by a hydraulic cylinder operated through the control system.

The exhaust hood assembly receives the exhaust gases from the turbine and directs them to the exhaust duct. Essentially, the exhaust hood assembly consists of the fabricated exhaust hood frame and the sheet metal exhaust hood skin. The exhaust hood frame consists of an outer barrel which is bolted to the after vertical flange of the turbine shell. An

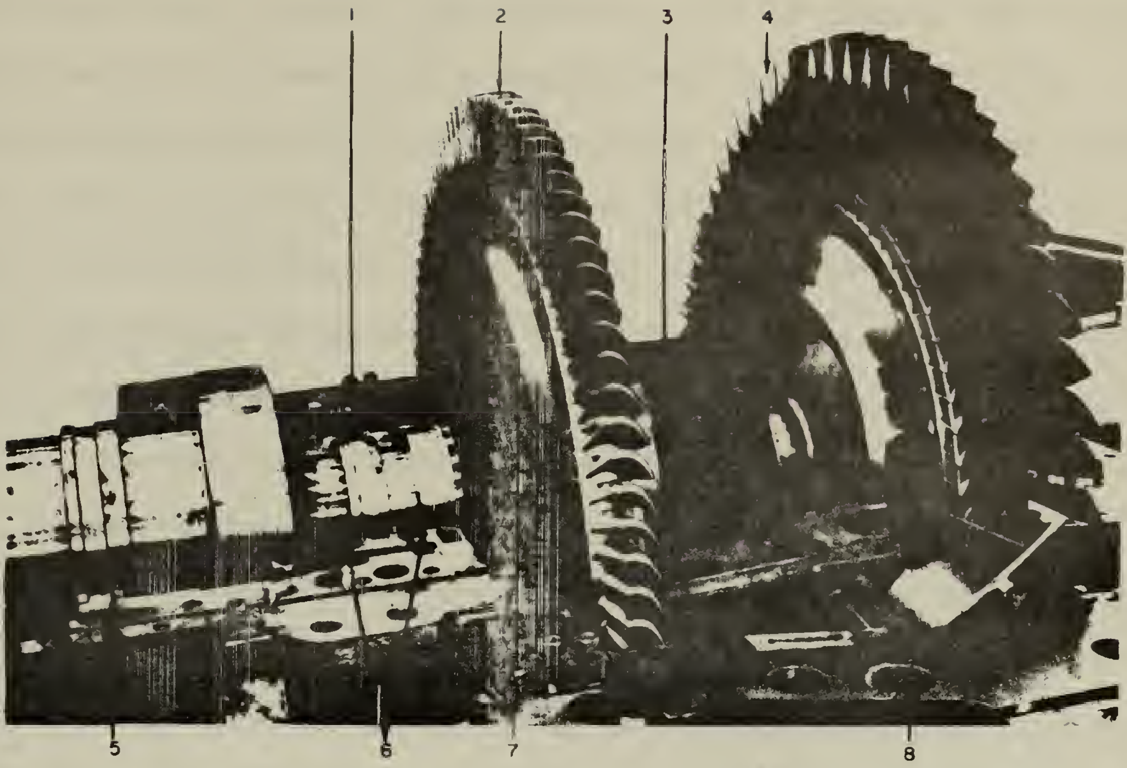
annular exhaust diffuser is located between the inner and outer barrels. Turning vanes are located at the end of the diffuser to turn the gases radially outward.

Marinized General Electric gas turbines seem suited to operate in a marine environment. The inlet guide vanes of 316 L (Low Carbon) Stainless steel were first used in 1961. All 75 units that have operated in a salt air environment have had no failures with them as opposed to the 304 series stainless steel which preceeded them. The turbine wheels are made of A286 high nickel content austenitic alloy with good fracture toughness, no low temperature ductility problem and very resistant to corrosion.

4. Auxiliary Systems

a. Bearings, Lubrication System, and Cooling Water System--

There are four gas turbine unit main bearings numbered 1-4 starting at the compressor end. All main turbine bearings are pressure lubricated by lube oil supplied from the turbine lube oil header. Branch lines carry the oil from the header to inlet ports located on the unit near the bearing to which the oil is supplied. Each bearing housing contains a passage which admits the lubricating oil from the feed line into an annulus around the bearing liner. From the liner, the oil goes directly to the journal and then drains back to the lube oil tank. There are also oil slingers which consist of steps in the shaft, on either side of the journal, that sling oil radially outward into the bearing cavity. Oil on the surface of the shaft which passes by the slingers is prevented from escaping by the oil seals, which are essentially labyrinth packings of oil deflectors. Pressure sealing air is admitted to these seals to minimize leakage of oil. (See Figure C. 11)(24.)



1. No. 3 bearing
2. High pressure (first stage) turbine
3. Diaphragm
4. Low pressure (second stage) turbine
5. Air seal
6. Oil deflector and air seal
7. First stage nozzle
8. Second stage nozzle

No. 3 bearing assembly (Typical).

Figure C.11 (24)

[illegible]

[illegible]

[illegible]

The lube oil system in general consists of a closed forced feed system including pumps, coolers, filters, valves and miscellaneous control and protective devices. The lube oil system and lube oil tank are integral parts of the gas turbine system. Taking suction on the 1700 gallon lube oil sump are three lube oil pumps: 1) main pump, driven by the accessory gear, 2) auxiliary pump, driven vertical a-c motor, and 3) an emergency pump driven by a vertical d-c motor. Lube oil coolers are installed in the lube oil tank with water supplied by the ship's cooling water system. A regulating valve opens and closes depending upon the oil temperature in the lube oil header, which attempts to maintain the oil temperature at 130° F.

b. Starting System, Turning Gear and Shaft Brake--

The gas turbine is started by a 300 horsepower, 3600 RPM induction motor in conjunction with a "twin disc" torque converter and step-up gear. The gear and torque converter are integral with the motor and connected to the gas turbine by a jaw clutch through the accessory gear. The motor will be supplied with power from the 1000KW diesel generator during inport periods, or whenever the main turbine is secured.

The propulsion shaft turning gear is mounted directly to the double reduction gear. It consists of a 15 horsepower reversible motor, operating on 440 volts, three phase, 60 cycle power. This will turn the propeller shaft approximately one revolution in ten minutes.

Since this gas turbine plant has an independent load turbine and a mechanical drive type of transmission, there will be sufficient torque at idle conditions to rotate the propeller. A shaft brake can be used to prevent the lineshaft from rotation. The brake is of the disc clamp type, hydraulically operated and mounted near the reduction gear. It will be

designed to absorb the kinetic energy of the propeller (neutral pitch), line shafting, main reduction gear and the power turbine with the ship dead in the water and with the gas turbine at idle fuel setting. Ten separate braking pads, mounted on a horseshoe-shaped bracket, are hydraulically operated and clamped to a rotating brake disc to produce braking action. No lubrication or cooling system is required, as heat is dissipated by radiation. The hydraulic pressure comes from the main C & RP propeller servo pump.

c. Base, Supports and Shock Requirements--

The gas turbine is mounted on a structural steel base. Fabricated as an integral part of the base is the oil tank which contains the lube oil supply for the gas turbine. On the aft end of the base is an oil sump, which receives oil from the No. 3 and 4 bearings. The oil sump is connected to the oil tank by an oil channel which is fabricated along the web of one of the longitudinal I-beam members of the structural steel base. Finished pads are provided on the bottom of the base to facilitate the mounting of the unit on the foundation sole plates.

The gas turbine supports consist of a flex plate just aft of the inlet hood, bolted to the inlet casing and welded to the base, with a similar flex plate at the aft end. The flex plates are rigid laterally to keep athwartship alignment, but are flexible axially to allow for thermal expansion. The aft flex plate can be shimmed if required to align the unit to the load gear. A jacking pad is provided under the turbine shell to support the unit during disassembly, or removal of the load turbine.

APPENDIX D. LIST OF EQUIPMENT AND SPECIFICATIONS (24)

D.1 MAIN AND AUXILIARY MACHINERY

Item No.	Item	No./ Ship	Description
1.	Main Propulsion Gas Turbine	2	General Electric Model 5322 B single cycle, two shaft gas turbines. Complete with electric motor starters, unit base mounted and arranged for vertical inlet and exhaust ducts.
1A.		2 Sets	Speedtronic Governing and Temperature Control Systems.
1B.		2 Sets	Protection System consisting of overtemperature, overspeed, flame detectors, vibration detectors and alarm and trips.
1C.		2 Sets	Sequential System to provide logic functions required for start-up, running and shutdown.
1D.		2 Sets	Power Supply Systems a. Power output for inductor alternator on turbine shaft. b. 125 volt, dc power from ship's batteries - ac power obtained through a static converter.
1E.		2	Ignition Systems
1F.		2 Sets	Thermocouples

Item No.	Item	No./ Ship	Description
2.	Inlet Air System	2	Equipment includes: a. Air inlet house-weather protected, complete with demistors. b. Inlet silencer assembly with 8-foot (2.44 meter) length of silencing section. c. Transition duct between silencer and compressor. d. Dynavane-type separator after intercooler. e. Inlet ducts between silencer and filter house.
3.	Exhaust System	2	Equipment includes: a. Exhaust transition duct between HRSG and turbine. b. Expansion joint. c. 10' x 10' HRSG bypass duct with butterfly damper. d. Exhaust duct from HRSG.
4.	Supercharger Steam Turbine	2	12-stage impulse, single casing, Marine-type, for steam conditions of 450 psig, 750° F., 2" hg Abs. Full power steam rate of 6.95 lb/shp-hr rated hp (7750) at 4950 rpm.
5.	Reduction Gear	2	Marine type, articulated, double reduction gear. Single pinion input, single output, rated 50, 200 shp and 33.4:1 reduction ratio to 140 prpm. Complete with turning gear and shaft lock.
6.	Supercharger Compressor	2	Two-stage axial flow, maximum pressure ratio 1.4:1, 4950 rpm at full power, 348.5 lbs/sec, flow area 8.13 ft^2.
7.	Controllable and Reversible pitch Propeller		Capable of absorbing 50,200 shp, 140 rpm, 4-bladed, 22.3 ft. dia., P/D 21.031 of $K_aM_eW_a$ design as in Table 3.5.

Item No.	Item	No./ Ship	Description
8.	Heat Recovery Steam Generator (HRSG)	2	Unfired type to operate on gas turbine exhaust gas. Rated 1,253,000 lb/hr at 475 psig, 750° F. at non-return valve. Also recirculates 96,000 lb/hr heating water from the economizer and 15,000 lb/hr of saturated steam for ship's services.
9.	Deaerating Condenser	2	Scoop salt water circulated, single pass, straight tube type. To condense 117,000 lb/hr of 90% quality steam at 2" Hg. Abs. 14,500 GPM, total cooling surface 4000 ft^2 (See Table 3.7 and Appendix A.5)
10.	Condensate Pumps	4	Motor-driven rated 25 shp, 440V, 3-phase, 60 Hz. Capacity 400 gpm at feedwater temperature of 101° F. Total discharge pressure 60 psig.
11.	Intercooler	2	Scoop circulated, single pass continuous finned tube, air flow rate of 1,277,000 lb/hr, water flow rate 6,000 GPM, effectiveness, .95, maximum gas pressure drop .349 psia.
12.	Boiler Feed Pumps	4	Steam driven, rated 144.5 shp, with lube oil system. Capacity 500 gpm at temperature of 105° F. to total discharge of 700 psig. Sized for 100% capacity.
13.	Lube Oil System	2	Complete with pumps, valves, strainers and piping. To provide lube services to gas turbine, supercharger steam turbine and reduction gear. Mounted and packaged on separate skid.
14.	Shaft Brake	2	Disc clamp type, hydraulically actuated. Braking pads, mounted on a horseshoe-shaped bracket clamp to a rotating disc for braking. No lubrication or cooling system required.
15.	Auxiliary Boiler	2	Drum pressure 135 psig, 30,000 lbs/hr welded wall construction (see Table 3.8) Combustion Engineering V2M type.

Item No.	Item	No./ Ship	Description
16.	Cooling Water Pumps	4	Motor-driven, rated 78 shp, 440V, 3-phase, 60 Hz, sized at 60% capacity for 5820 gpm of cooling water at a total discharge head of 40 feet.
17.	Vacuum Pumps	2	Motor-driven 20 HP, 1170 RPM, Nash type, 50,000-250,000 lbs/hr steam rating.
18.	Steam Turbine Generators	4	1500KW rated, 450V, 3-phase, 60 Hz, 0.8 p.f., steam conditions 450 psig 2" Hg. Abs. Base mounted, complete with lubrication system.
19.	Motor Group Control	2	Grouped motor-starters for gas turbine auxiliaries. Drip-proof fully gasketed with pan-type dripshield.
20.	Main and Emergency Switchboards	1 Set	Drip-proof enclosure for distribution breakers for shipboard electrical service.
21.	Fuel System (Base mounted on gas turbine)	2	Distillate oil fuel system with high pressure air atomizing including: (2) Fuel oil inlet pressure gauges--line mounted. (2) Fuel oil pressure gauges. (2) Fuel oil stop valves. (2) Shaft-driven, variable displacement fuel oil pump hydraulically actuated. (2) Sets of fuel nozzles. (2) Fuel oil flow dividers. (2) Fuel oil filters ahead of stop valves. (2) Fuel oil filters ahead of flow dividers. (2) Atomizing air compressor, shaft-driven. (2) Starting atomizing air compressor, motor-driven. (2) Centrifugal purifiers.
22.	Central Operations System	2	Consisting of: (2) Engine room consoles with

Item No.	Item	No./ Ship	Description
			information display, monitor and alarm for continuous automatic surveillance of operating conditions. Program controls, sub-loop controls, data logger (optional), bell logger, start-stop of gas turbine, engine order telegraph, phone, throttle control, sensors and instruments.
			(1) Bridge Operating Consoles with throttle control, engine order telegraph, phone, and supervisory instruments.

APPENDIX E. PROPOSED PLANT LAYOUT FOR AOE-1 CLASS INCLUDING MACHINERY SIZE AND WEIGHTS

TABLE E.1--WEIGHTS OF MAJOR COMPONENTS (PER ENGINE ROOM) (24)(42)

1.	MS 5322 Gas Turbine and Accessories	235,000 lbs.
2.	Supercharger Compressor and Steam Turbine	85,000 lbs.
3.	Intercooler	48,000 lbs.
4.	Main Condenser	28,000 lbs.
5.	Main Reduction Gear	247,000 lbs.
6.	Controllable and Reversible Pitch Propeller	136,700 lbs.
7.	Heat Recover Steam Generator (Wet)	210,000 lbs.
8.	1500 KW Turbogenerators (2)	60,000 lbs.
9.	Main Feed Pumps (2)	14,000 lbs.
10.	Main Condensate Pumps (2)	6,000 lbs.
11.	Vacuum Pumps (2)	8,200 lbs.
12.	Auxiliary Boiler (Estimated Wet)	95,000 lbs.

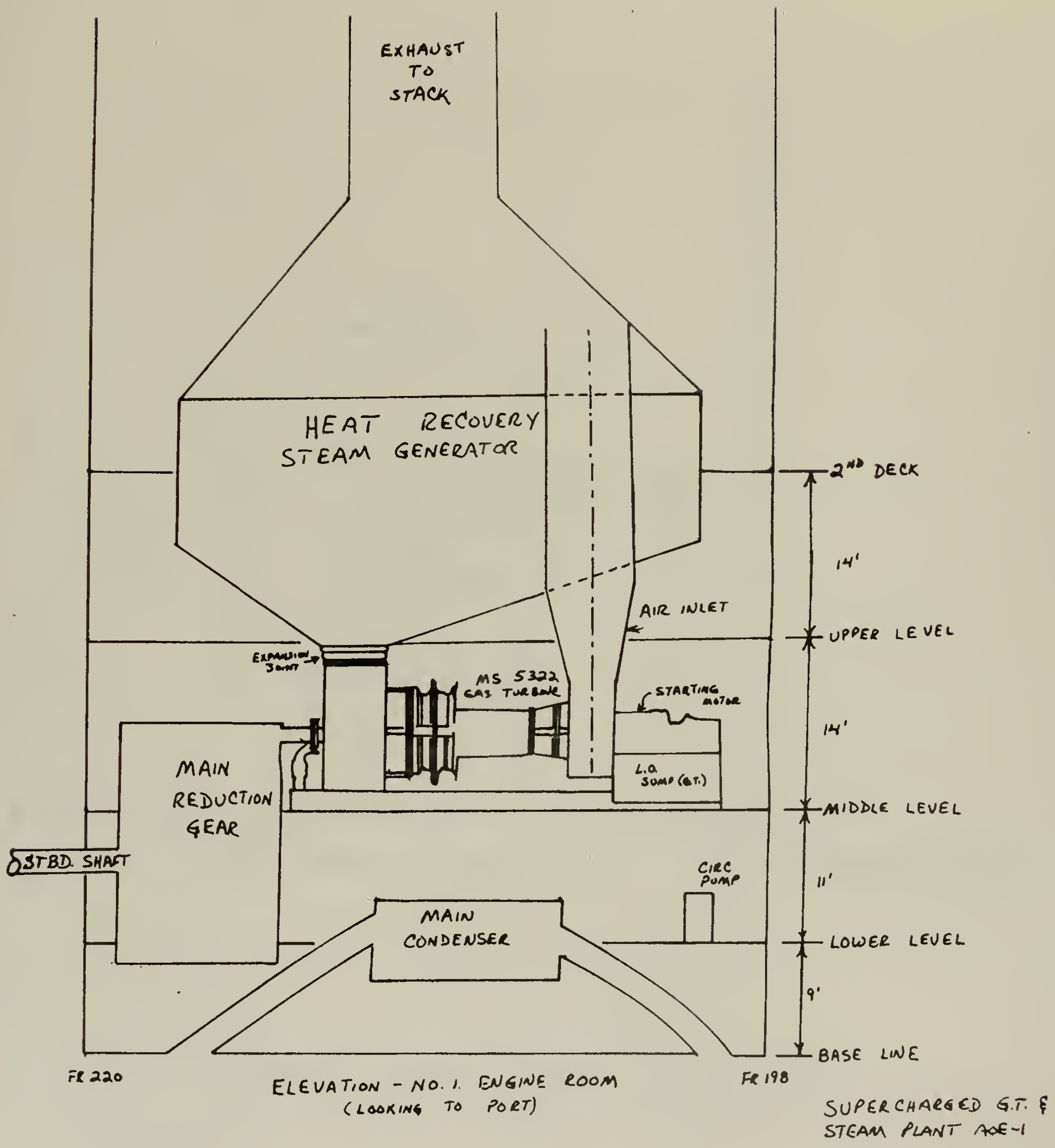


Figure E.1 Elevation--No. 1 Engine Room

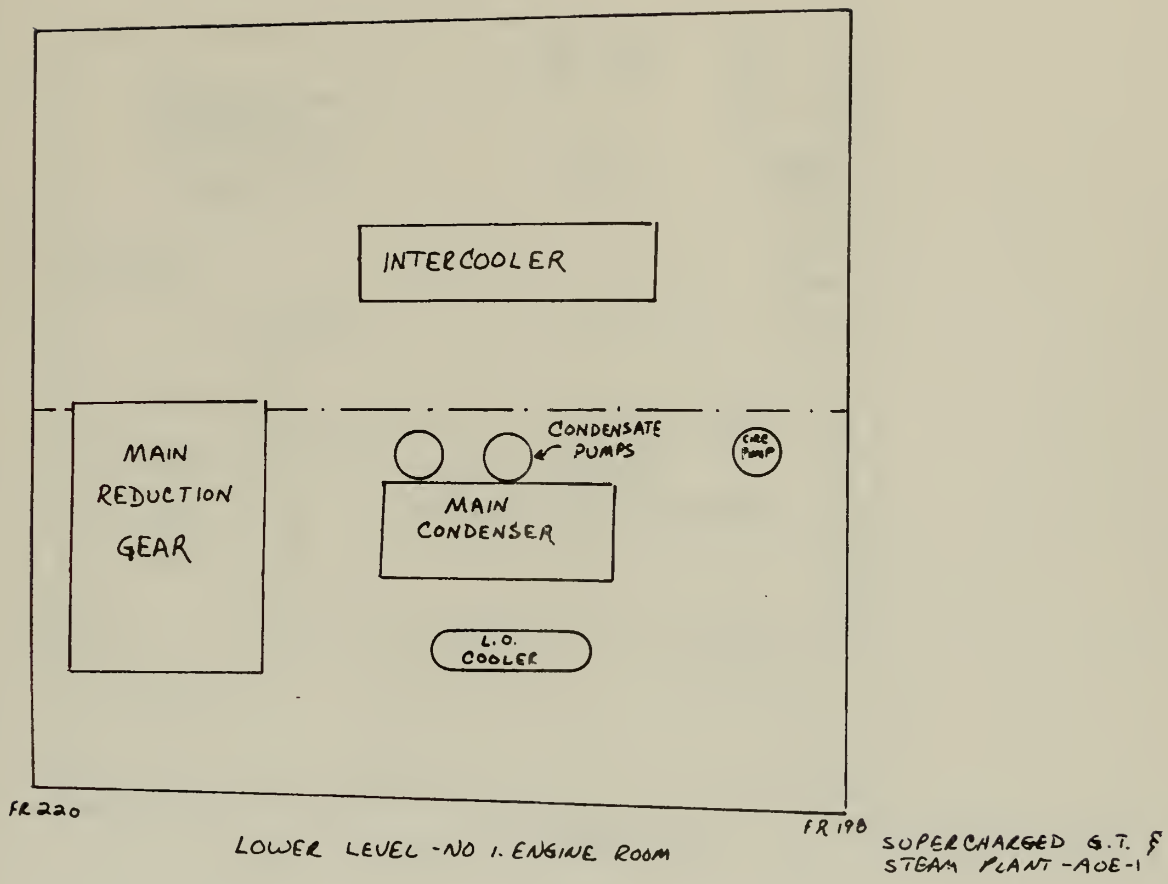


Figure E.2 Lower Level--No. 1 Engine Room

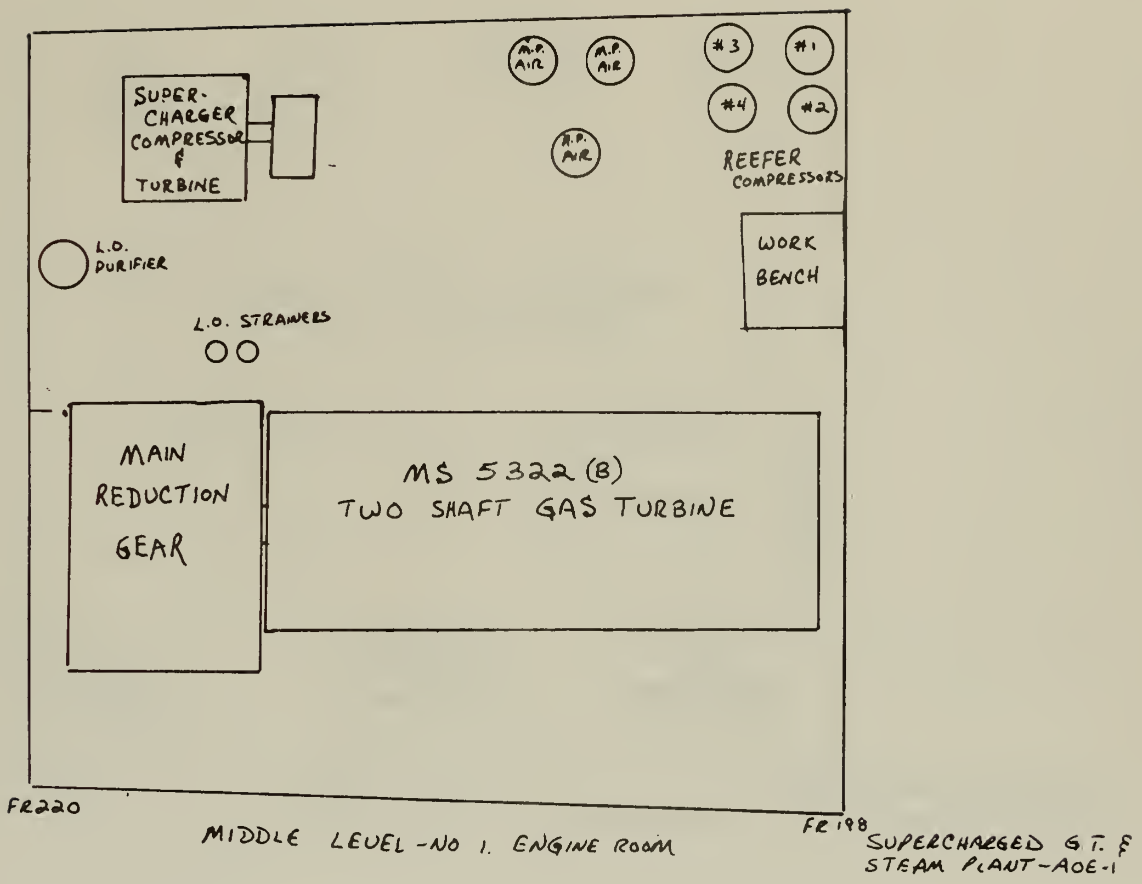

M.P. AIR
M.P. AIR
H.P. AIR
#3
#1
#4
#2
REEFER COMPRESSORS
SUPER-CHARGER COMPRESSOR & TURBINE
L.O. PURIFIER
WORK BENCH
L.O. STRAINERS
MAIN REDUCTION GEAR
MS 5322 (B) TWO SHAFT GAS TURBINE
FR220
FR 198
MIDDLE LEVEL - NO 1. ENGINE ROOM
SUPERCHARGED G.T. & STEAM PLANT - AOE-1

Figure E.3 Middle Level--No. 1 Engine Room

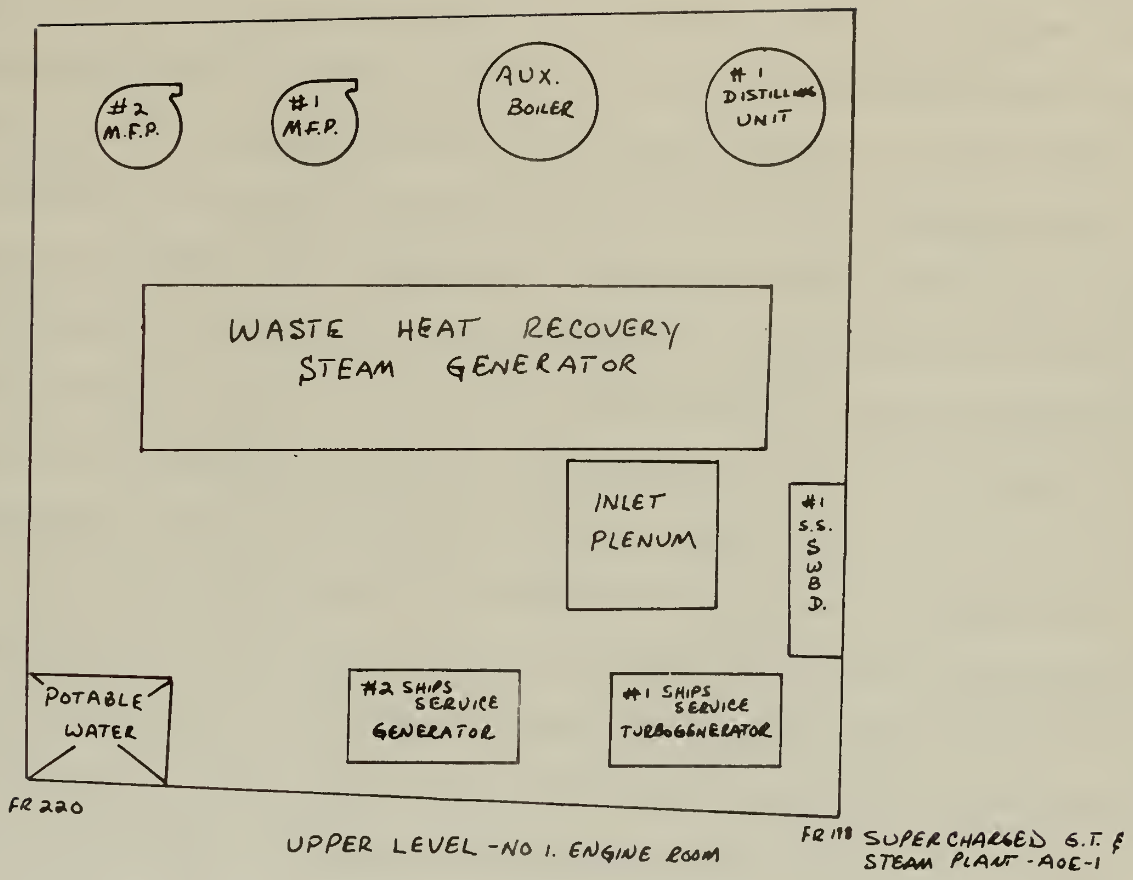

#2 M.F.P.
#1 M.F.P.
AUX. BOILER
#1 DISTILLING UNIT
WASTE HEAT RECOVERY STEAM GENERATOR
INLET PLENUM
#1 S.S. SWBD.
POTABLE WATER
#2 SHIPS SERVICE GENERATOR
#1 SHIPS SERVICE TURBOGENERATOR
FR 220
FR 198
UPPER LEVEL -NO 1. ENGINE ROOM
SUPERCHARGED G.T. & STEAM PLANT -AOE-1

Figure E.4 Upper Level--No. 1 Engine Room

APPENDIX F--SUGGESTED MANNING SCHEDULE

1. Introduction

A manning schedule is proposed as a possible example of what the AOE-1 Class of ship would require with this type of power plant. While gas turbine manufacturers in general indicate that gas turbine plants may be run with essentially unmanned spaces, for practical purposes, the Navy would be reluctant to do this, as the mission of a Navy ship is different than that of a merchant ship. Additionally, as stated before, most of the large maintenance (up to but not including compressor reblading) would be done by Navy personnel, thus there is needed a large reserve of talent aboard ship. These men can man the main spaces during General Quarters and Underway Replenishment, which is the primary function of this ship.

Two schedules are shown for operation of this plant. One shows the basic propulsion personnel who would be assigned to the ship, the other is a sample breakdown of the normal cruising watch, which would allow four or five sections. In all cases, control is generally expected to remain with the bridge console. Telephone communication would be continuous between the pilot house operator and the 'top watch' in No. 1 engine room, commonly called main control. The number of personnel in the engine rooms would remain constant whether control is on the bridge or in the engine room.

2. Suggested Schedule

a. Propulsion System Personnel

		LOCATION		
Number & Grade		General Quarters	Sea Details & U/W Replenishment	Cruise
1-LCDR	Engineer Off.	No. 1 Engine Room	No. 1 Engine Room	----

		LOCATION		
Number & Grade		General Quarters	Sea Details & U/W Replenishment	Cruise
1-LT	Main Propulsion Assistant	No. 2 Engine Room	No. 2 Engine Room	EOOW
1-LT	Gas Turbine Assistant	No. 1 Engine Room	No. 1 Engine Room	EOOW
1-LTJG	Damage Control Assistant	Damage Control Central		EOOW
1-E-9	Chief Engineman	No. 1 Engine Room	No. 1 Engine Room	EOOW
1-E-7	Chief Machinist's Mate	No. 2 Engine Room	No. 2 Engine Room	EOOW
1-E-7	Chief Boilerman	No. 1 Engine Room	No. 1 Engine Room	EOOW
1-E-7	Chief Internal Comm. Tech.	Bridge Console	Bridge Console	----
2-E-6	Engineman First Class	No. 1, No. 2 Engine Rooms	No. 1, No. 2 Engine Rooms	Top Watch
1-E-6	Electrician First Class	No. 1 Switchboard	No. 1 Switchboard	Swbd.
2-E-5	Machinist's Mate Second Class	No. 1, No. 2 Engine Rooms	No. 1, No. 2 Engine Rooms	Prop. Control
2-E-5	Boilerman Second Class	No. 1, No. 2 Engine Rooms	No. 1, No. 2 Engine Rooms	Upper Level
4-E-4	Machinist's Mate Third Class or Engineman Third Class	No. 1, No. 2 Engine Rooms	No. 1, No. 2 Engine Rooms	Lower Level
2-E-5	Electrician Second Class	No. 2 Switchboard	No. 2 Switchboard	Swbd.
1-E-5	Internal Comm. Technician Second Class	I.C. Room	I.C. Room	I.C. Room
6-E-3	Fireman	No. 1, No. 2 Engine Rooms	No. 1, No. 2 Engine Rooms	Lower Level
28 TOTAL				

b. Underway Cruising Condition--Personnel per Engine Room

1 - Engineering Officer of the Watch (EOOW) (Main control only)

1 - Engineman First Class (Top Watch)

1 - Engineman, Machinist's Mate or
Boilerman Second Class (Controls)

1 - Electrician or Internal Communications
Technician Second Class (Switchboards)

1 - Fireman

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